AN INVERTED BRAYTON CYCLE
APPLICATION TO NAVAL MARINE
GAS TURBINES

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ABSTRACT

A thermodynamic analysis is conducted for the Allison gas turbine used for electrical-power generation on the Spruance-Class destroyer. The addition of an induced-draft fan of the proper pressure ratio results in a 9% gain in gas-turbine power and efficiency.

A method for analyzing the cost-effectiveness of any system on a naval ship is presented. The ship-impact cost was found to be $482,525 for adding the inverted Brayton cycle to the Spruance-Class destroyer. This was balanced against the option of life-cycle fuel savings of $1,057,440 or increased electrical power of 540 kw. Although this represents a cost-effective addition, the value to the naval ship designer is small because the magnitude of the savings is only 6% of the total fuel for electrical-power generation.

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TABLE OF CONTENTS

LISTING OF FIGURES

LISTING OF TABLES

NOMENCLATURE

A	Flow Area
a	Speed of Sound
bbl	Barrel
C	Absolute Velocity
C_θ	Absolute Tangential Velocity
C_D	Drag Coefficient
c_p	Specific Heat at Constant Pressure
EHP	Effective Horsepower
h	Enthalpy
Δh	Change in Enthalpy
h_{is}	Isentropic Enthalpy
hp	Horsepower
LHV	Lower Heating Value
M	Mach Number
$\dot{m}_a$	Mass Flow Rate of Air
$\dot{m}_f$	Mass Flow Rate of Fuel
$\dot{m}_s$	Mass Flow Rate of Steam
$\dot{m}_{total}$	Mass Flow Rate of Air and Fuel
MW	Molecular Weight
N	Revolutions per Minute
P	Absolute Pressure
P_r	Relative Pressure
PR	Pressure Ratio
$\dot{Q}_H$	Heat-Energy Input
R	Universal Gas Constant

r_h	Radius at Rotor Hub
r_m	Radius at Mid-Rotor
r_t	Radius at Rotor Tip
S	Wetted Surface
s	Entropy
s_{is}	Isentropic Entropy
SFC	Specific Fuel Consumption
SHP	Shaft Horsepower
T_o	Stagnation Temperature
ΔT	Change in Temperature
TPI	Tons per Inch Immersion
u	Peripheral Velocity
V	Velocity
v	Specific Volume
W	Relative Velocity
W_θ	Relative Tangential Velocity
$\dot{W}_c$	Compressor Work
$\dot{W}_{idf}$	Induced-Draft Fan Work
$\dot{W}_t$	Turbine Work
x_{is}	Isentropic Quality
α	Absolute Flow Angle
β	Relative Flow Angle
γ	c_p / c_v
η_{is_c}	Compressor Isentropic Efficiency
η_{is_t}	Turbine Isentropic Efficiency

η_p	Polytropic Efficiency
η_{th}	Thermal Efficiency
ρ_o	Stagnation Density
ϕ	Work Coefficient
ω	Angular Velocity

CHAPTER 1 INTRODUCTION

1.1. BACKGROUND

The rising price of fuel has been one of the important factors in the escalating costs of operating our naval ships in recent years. This higher cost of fuel combined with the shortage of domestic reserves has led to immediate cutbacks in ship operations in order to stay within operating budgets and to conserve petroleum reserves. Decreasing the rate of fuel consumption aboard ship is further complicated by the increasing demands for higher power generation from shipboard machinery systems found in the evolution of naval ship design recently. The most significant increases are in propulsion and particularly in electric power plants as shown by Graham [9]*. Because the naval ship designer cannot solve the energy crisis, he is challenged with optimizing thermal efficiency and decreasing fuel consumption in the navy's non-nuclear fleet.

The aircraft-derivative marinized gas turbine is projected as the primary non-nuclear main-propulsion power plant for U. S. Navy ships in the remaining quarter of this century. The development of the gas turbine into an efficient and versatile prime mover has resulted from recent component improvements which in turn were the products of metallurgic, aerodynamic, and heat-transfer

*Numbers in brackets, [ ] , refer to Bibliography

advances. The gas turbine is attractive to the ship designer because of its low weight per shaft horsepower, rapid response to power changes, lower manning levels and maintenance requirements. Additionally, the gas turbine can be removed modularly for overhaul and is readily adaptable to automatic control. But the gas turbine needs to advance its state of the art in several areas including fuel consumption in order to remain competitive with alternative propulsion systems. Leopold [15] reports that the specific fuel consumption (sfc = lbm fuel/SHP -hr) for a gas-turbine propulsion plant was significantly higher than a steam or diesel plant (0.845 to 0.761 and 0.560) at endurance speed in a recent ship-design trade-off study. These typical results from comparing various propulsion plants demonstrate the dilemma facing the ship designer. The advantages of the gas turbine are important to the designer, but its poor fuel-consumption rate is accented by increasing fuel costs and fuel shortages resulting from the energy crisis. Clearly, if the gas turbine is to become the propulsion plant of the future, its specific fuel consumption must be improved.

The U. S. Navy has undertaken many studies to improve the specific fuel consumption of the gas turbines. A supercharger cycle investigated by Cuthbert [5] gave

a large gain in horsepower at the same fuel rate but with a penalty in weight and volume for the engine. Another study for the Navy [7] investigated advanced marine concepts such as waste-heat recovery, water/steam injection, combined plant/cycle and closed-cycle systems and reported mild improvements in sfc. Bratton, McLean, and van Reuth [20] investigated ceramic gas-turbine technology and report increased turbine-inlet temperatures in the range of 2500°F can be reached with only the first-stage-stator vanes using ceramics. The corresponding 25% improvement in sfc put the gas turbine in the same range as typical diesel engines.

Other efforts to reduce shipboard fuel consumption are reflected in current ship designs which integrate the ship's auxiliary or propulsion systems to take advantage of waste heat. The FFG 7 Class design incorporates waste-heat recovery from the cooling system of the ship's-service diesel generator by a secondary hot-water circulating system to provide the heating requirements for the fuel and lubricating-oil systems, potable-water system, missile-launcher anti-icing system, and the distilling plants. Gas-turbine exhaust-heat-recovery boilers are being used on the DD 963 Class to supply auxiliary steam for the heating of fuel, lubricating oil and potable water, for the operation of

the distilling plant, laundry and commissary and for the de-icing of the missile launcher. The Sea Control Ship design utilized a waste-heat-recovery system similar to that of the DD 963 Class to provide steam requirements for the operation of the distilling plant, laundry and commissary, and heating of potable water. These integrated systems all take advantage of excess energy developed by gas turbines and therefore decrease the total requirements for fuel consumption in the ship.

Another effort to utilize energy developed but not used by the gas-turbine cycle is the COGAS propulsion system. In this integrated system, gas-turbine exhaust normally discharged from the exhaust stack to atmosphere is passed through a waste-heat-recovery unit. The "boiler" tube surfaces in the recovery unit transfer this exhaust-gas heat into steam which is used to drive a propulsion steam turbine. Both gas turbine and steam turbine are coupled to a common reduction gear. Overall cycle efficiency is improved because the steam is generated entirely in an unfired boiler located within the gas-turbine exhaust stack. Abbott [2] reports an increase of 7000 hp using this cycle with no increase in fuel consumption. The result is an impressive 30% decrease in specific fuel consumption.

A performance penalty must normally be accepted when

one adds an exhaust-heated boiler to a gas turbine as has been proposed with the COGAS propulsion plant and the integrated auxiliary systems. This penalty is often expressed as a pressure loss (inches of H_2O) similar to exhaust or intake stack losses. The additional pressure drop caused by the boiler could typically be expected to be in the range of 3-8% of the total power for medium-sized gas turbines.

Wilson and Dunteman reported in Reference 6 that a Canadian purchaser of a Ruston and Hornsby TA 1250 kw gas turbine had written to the manufacturers to inform them that their performance penalty seemed to be in error. The reason for his statement was that he had decided to add a waste-heat boiler to his existing TA turbine, but he could not allow the power output to decrease by the 100 hp which was predicted. He, therefore, decided to install an electrically driven induced-draft fan after the waste-heat boiler so that the pressure at the turbine-exhaust flange would be restored to atmospheric. He found that he could obtain the required gas-turbine power increase of 100 hp with a fan absorbing approximately 60 hp. It was not obvious why an expenditure of 60 hp would result in a gain of 100 hp.

One way of explaining this phenomenon is that since the exhaust gas had been cooled by the heat-recovery unit,

the fan had to compress a smaller volume of gas than it would have at the higher gas temperature at the exhaust flange of the turbine. Whereas the normal Brayton cycle (gas-turbine cycle) consists of compression-combustion-expansion, the induced-draft fan had the effect of inverting the Brayton cycle at the end of the gas-turbine cycle, resulting in expansion-cooling-compression. This inverted Brayton cycle utilized the existing turbine and heat exchanger in the case cited above and recovered the gas-turbine power which was lost when the waste-heat-recovery unit was added. One might then expect a larger induced-draft fan to add additional power to the gas turbine, above the initial power rating, since the same thermodynamic principles are involved.

Such an inverted Brayton cycle application for industrial-power-generation gas turbines was investigated by Wilson and Dunteman [6]. Their results showed a return on investment of 30% and a net power gain of 100 hp for the inverted cycle considered as an add-on device for a standard power-producing gas turbine. This large return on investment was calculated on the assumption that the gas turbine would be exhausting into the waste-heat boiler in any event, and would suffer a power loss as a result of the back pressure. Accordingly, the only capital investment necessary was for the induced-draft fan.

Another inverted Brayton cycle study was undertaken and reported by Gasparovic [8]. His primary interest was in heavy-duty gas turbines suitable for use on merchant ships. He found improvements in cycle efficiency around 20% for a typical heavy-duty gas turbine. The same gains were found for the specific work of the gas turbines. These were significant findings when one considers the long periods of time merchant ships spend underway annually. Since efficiency and fuel consumption are related by an inverse relationship, the increase in cycle efficiency could be interpreted as fuel savings.

Gas turbines and integrated waste-heat cycles appear to be the wave of the future for naval propulsion and electrical-power generation. The inverted Brayton cycle has shown significant prospects for improving the efficiency of the gas-turbine cycle for industrial and merchant ship application. Since the naval ship designer is troubled by the high specific-fuel-consumption rate with the current aircraft-derivative marine gas turbines, an investigation into the potential of applying an inverted Brayton cycle to naval ships appears worthwhile.

1.2. GOAL OF INVESTIGATION

The reduction of fuel consumption for a particular piece of equipment is not a sufficient goal in machinery

design for ship applications. If the method of fuel savings results in increased machinery weight or volume, or increased electrical load, the ship design may have to grow even more than the machinery in order to maintain constant performance. For example, a fuel-savings method for generating compressed air might require a larger, heavier compressor. The savings in compressed-air fuel consumption might then be offset by increased fuel consumption in propulsion to drive the now larger and heavier ship the same distance. Combining the goal of reducing fuel consumption for machinery and the goal of minimizing the ship design may not be compatible if the impact of the system adversely affects the total-ship system.

The goal of this study is to conduct a thermodynamic analysis of the inverted Brayton cycle in order to investigate the performance, efficiency and fuel-consumption effects on naval marine gas turbines and then to measure the impact of the cycle components with the anticipated gains in performance in relation to the ship as a system.

1.3. SUMMARY OF APPROACH

The two major goals of this study discussed in the preceding section lend themselves to two distinct areas of investigation. First, the thermodynamic analysis of

the inverted Brayton cycle application to naval marine gas turbines will be undertaken to determine the effect on efficiency, power, and fuel consumption. The second area of the study, measuring the impact of the cycle components against the performance in a total ship-system environment, will follow.

The analysis of the thermodynamic cycles found in Chapter 2 has three separate sections. The analytical approach for the Brayton cycle, Brayton cycle with waste-heat boiler, and inverted Brayton cycle are described.

A 9 percent gain in power and efficiency is achieved from adding an induced-draft fan to the cycle which generates electrical power. The failure to achieve similar gains in the propulsive cycle is included in the study to demonstrate that the inverted Brayton cycle should not be applied randomly to any gas turbine by the ship designer.

A method for analyzing naval ship systems is presented in Chapter 3. This method utilizes the concept of a total-ship system and is valid for measuring the cost-effectiveness of any subsystem. Various alternatives using the power and efficiency gains from the inverted Brayton cycle were investigated using this method for analyzing naval ship systems. Despite the promising thermodynamic gains in power and efficiency, the various

applications on naval ships were found to be only marginally cost-effective, at best.

Conclusions and recommendations are found in Chapter 4. This includes cycle uses which do not warrant further investigation, as well as those which showed promising gains through efficiency or power improvements. Finally, areas requiring further study will be discussed.

A balance in the study was intended between calculating the thermodynamics and measuring the cost-effectiveness. In this regard, simplifying assumptions have been made in each section in order to provide a broad view of the total problem with enough depth and accuracy to produce valid results in any and all of the sub-sections. By using this approach, the potential of the inverted Brayton cycle can be fully investigated in naval ship applications.

CHAPTER 2 THERMODYNAMIC ANALYSIS

2.1. INTRODUCTION

The purpose of conducting the thermodynamic analysis is to calculate the power and efficiency of the inverted Brayton cycle. In order to maintain a consistent comparison, the Brayton cycle and the Brayton cycle with the waste-heat boiler are also analyzed and the components necessary to complete the cycles are investigated. The calculations are completed for an electrical-power system with the result of a 9 percent gain in power. The same procedure is used for a propulsion system to demonstrate the failure of this cycle to increase in total power.

This study limits the thermodynamic analysis to one climatic condition and one gas-turbine load condition. The objective in this approach is to investigate the potential of the inverted Brayton cycle to determine which applications should be investigated with the added detail of varying climatic and load conditions. The standard gas-turbine design conditions of full power and 100°F atmospheric temperature are used.

2.2. BASELINE SHIP

Two separate gas turbines are analyzed, one for ship's propulsion and another for electrical-power generation.

A summary of the characteristics of two gas turbines being introduced into the Navy for propulsion and electrical-power generation is found in Table 1. A description of their cycles on their particular ships follows.

The main-propulsion gas-turbine engine and cycle selected for study was the LM 2500 and COGAS system investigated for use by NAVSHIPS [7] on the DG/AEGIS and by Abbott [2]. This configuration uses the exhaust heat from the LM2500 gas turbine to generate steam in a waste-heat-recovery unit (WHRU). This steam drives a propulsion steam turbine in parallel with the gas turbine. A common gearbox connects both prime movers to the propeller shaft. There is one LM2500 and one steam turbine for each of the two propeller shafts.

The electrical-power-generation gas turbine selected was the one used in the Spruance Class (DD 963) destroyer. The electric plant and auxiliary system was described for NAVSHIPS by Whiteford [21]. Three 2,000-kw, 440-volt, 60-Hz ship's-service and emergency generators are driven by Allison gas-turbine engines. Each ship service/emergency generator gas turbine will discharge through a waste-heat boiler without a bypass arrangement. The amount of steam generated will vary with turbine electric load, ambient air temperature, and other

TABLE 1

GAS-TURBINE CHARACTERISTICS

GAS TURBINE	LM 2500	ALLISON
Application	Propulsion	Electrical Power
Rated Power	21,500 hp	3,640 hp
Ambient Temperature	$100^{o}F$	$100^{o}F$
Turbine-Inlet Temperature	$2150^{o}F$	$1700^{o}F$
Mass Flow of Working Fluid	463,000 lbm/hr	109,370 lbm/hr
Fuel LHV	18,200	18,200
Weight	22 tons*	24.8 tons**
Length	267 in.	287 in.
Width	84 in.	82 in.
Height	84 in.	82 in.

* includes gas-turbine module
** includes generator

variables. The maximum output of each boiler is about 11,000 lbm/hr of 100 psi, saturated steam, and normal rating is 7,000 lbm/hr each. The excess steam which cannot be used by auxiliary services is automatically disposed of in an atmospheric condenser which is part of the boiler package.

The potential exists to develop the steam power from the propulsion system or from the electrical-generation system into additional electrical power through a steam turbine. This potential is not currently developed by naval ship designers for several reasons. In the case of the propulsion system, the demands of the electrical plant and propulsion plant are independent. If the electrical demand were high while the propulsion demand was low, as could happen in port, at anchor, or even in transit, there might be insufficient steam to meet the electrical load. Also, the propulsive load is subject to rapid changes in demand while the electrical load is more even. In the case of the electrical generation system, standard Navy practice calls for two generators to be in service simultaneously so that vital loads can be maintained in an emergency if one generator is lost. With two generators on the line, there is no need for additional power. Although the potential exists, this study will not integrate these

two systems for the reasons stated above and because the baseline ships kept the systems segregated. The purpose of the study is to investigate the potential of the inverted Brayton cycle. The potential of waste-heat systems has been well documented.

2.3 ANALYTICAL APPROACH

The objective of the analysis is to calculate the net power and efficiency of the inverted Brayton cycle and compare them to the Brayton cycle with the waste-heat boiler. The original Brayton cycle will also be calculated to indicate the accuracy of the approach. Since the propulsion cycle includes a steam turbine using waste heat as an energy source, the steam cycle will also be investigated. A preliminary design of an induced-draft fan will be accomplished for the electrical-power application to demonstrate feasibility of the design.

A summary of the conditions and flow rates of the two gas turbines is found in Table 2, as well as assumptions made to complete the calculations. This information, taken from another description of the same two gas turbines by Rains [17] and supplemented by data from MIT course 13.22, Propulsion Systems, and MIT Professional Summer, 1975 [16] allows calculation of all of the necessary state points in the cycles.

TABLE 2

STATES AND ASSUMPTIONS FOR THE GAS-TURBINE CYCLE

STATES

GAS TURBINE	LM 2500	ALLISON
Ambient Temperature	$100^{o}F$	$100^{o}F$
Turbine Inlet Temperature	$2150^{o}F$	$1700^{o}F$
Pressure Ratio	16.8	12.8
Intake Losses	4" H_2O	6" H_2O
Exhaust Losses	10" H_2O	10" H_2O
Mass Flow of Working Fluid	463,000 lbm/hr	109,370 lbm/hr
Fuel LHV	18,200	18,200

Reference: 16 & 17

ASSUMPTIONS

1. Molecular Weight of Working Fluid	28.9
2. Exhaust Gas	400% Theoretical Air
3. Compressor Isentropic Efficiency	85%
4. Turbine Isentropic Efficiency	86%
5. Induced Draft Fan Isentropic Efficiency	85%
6. Waste Heat Boiler Pressure Loss	6%
7. Combustor Pressure Loss	6%

2.3.1. THE BRAYTON CYCLE

The Brayton cycle is the basic gas-turbine cycle. An enthalpy-entropy diagram of this cycle, shown in Figure 1, displays the three basic steps of compression-combustion-expansion in the cycle, as well as some of the pressure losses. Following the cycle on the h-s diagram, intake losses are indicated by moving off the atmospheric pressure line shown as p_o. Compression is shown from p_1 to p_2 and this work into the cycle can be measured using an estimated isentropic efficiency. Combustion occurs from state 2 to 3 and a pressure loss will occur as shown by dropping below the line of peak pressure. Work out of the cycle is shown as expansion from state 3 to 4. Again, work out is found using an estimated isentropic efficiency. Finally, exhaust-stack pressure losses occur from state point 4 to atmospheric pressure, p_o.

The net work of the cycle is the difference between the turbine expansion (state 3 to 4) and the compression (state 1 to 2). Thermal efficiency is calculated as the net work out divided by the heat energy into the cycle (state 2 to 3).

2.3.2. THE BRAYTON CYCLE WITH WASTE HEAT BOILER

This cycle is the baseline for the studies. An enthalpy-entropy diagram for the cycle is found in

FIGURE 1
BRAYTON CYCLE

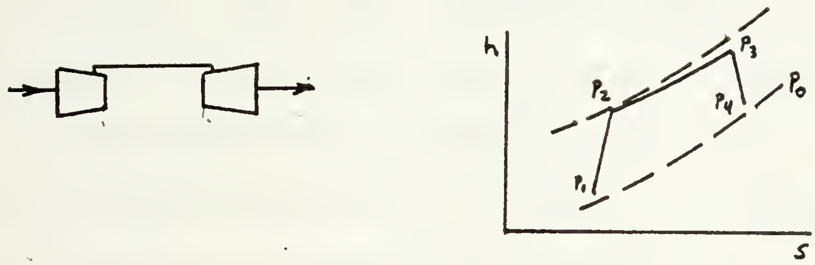


FIGURE 2
BRAYTON CYCLE
WITH WASTE-HEAT BOILER

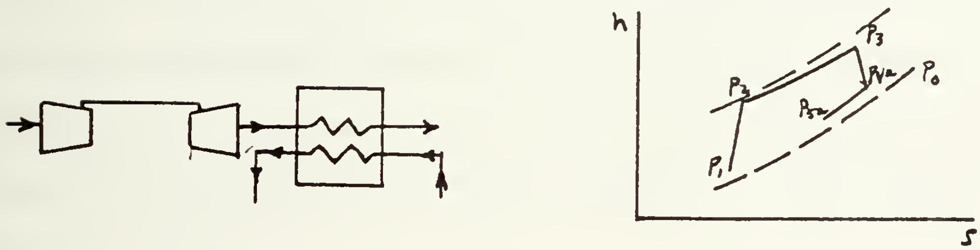


FIGURE 3
INVERTED BRAYTON CYCLE

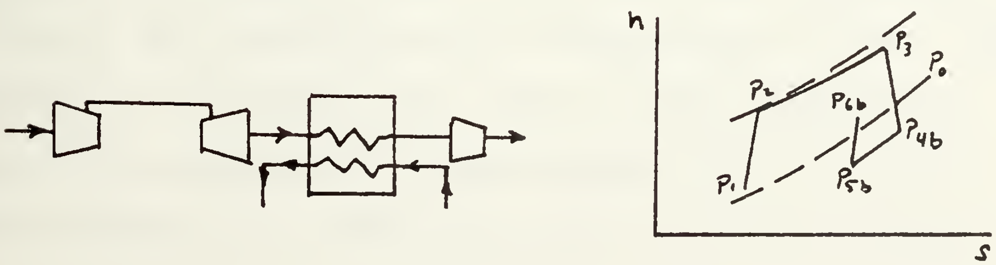


Figure 2. Intake losses (p_o to p_1) and compression (p_1 to p_2) are the same as the Brayton cycle without the waste-heat boiler. The first change in the cycle is the expansion-phase exit pressure (p_{4a}) which is higher than before. This higher exit pressure results in lower work out of the turbine (i.e.; $h_{4a}>h_4$) and is caused by a pressure loss of the exhaust gas through the waste-heat boiler, referred to earlier as backpressure. Flow through the waste-heat boiler is shown (state point 4_a to 5_a) where point 5_a indicates the same exhaust-stack pressure loss used in the Brayton cycle.

The net work of the cycle is the difference between the turbine expansion (state 3 to 4_a) and the compression (state 1 to 2). Thermal efficiency is calculated as the net work out divided by the heat energy into the cycle (state 2 to 3).

2.3.3. THE INVERTED BRAYTON CYCLE

The inverted Brayton cycle is shown added to the original Brayton cycle in the enthalpy-entropy diagram in Figure 3. Intake losses (p_o to p_1) and compression (p_1 to p_2) are the same as the two previous cycles. The pressure at the gas-turbine exit is below atmospheric (p_{4b}) and is determined by the pressure ratio of the induced-draft fan. This shows that there is a greater

enthalpy change across the turbine and therefore greater work out than in the previous cycles. The gas flow through the waste-heat boiler and the accompanying pressure loss is shown (state 4_b to 5_b). Additional work into the cycle is required to compress the gas to exhaust pressure as shown (state 5_b to 6_b). Again, exhaust-stack pressure losses complete the cycle to atmospheric pressure.

The work into this cycle is the original compression (state 1 to 2) and the added compression from the induced-draft fan (state 5_b to 6_b). The net work is the difference between the turbine expansion (state 3 to 4_b) and the work in. Thermal efficiency is calculated as the net work out divided by heat energy into the cycle (state 2 to 3).

2.3.4. STEAM CYCLE

The addition of the induced-draft fan to the Brayton cycle with the waste-heat boiler results in a different gas-turbine exhaust temperature. This temperature will affect the steam temperature in the waste-heat boiler which will change the power available from the steam turbine. It is not necessary to design the steam-cycle components since they are part of the baseline ship, but the changes in the steam-cycle power are critical to the cycle. A schematic diagram showing the components

and design points is found in Figure 4. Table 3 lists the information which is known for this portion of the problem as well as assumptions made to complete the analysis.

The effect of changing the exhaust-gas temperature can be calculated using the First Law of thermodynamics and solving simultaneous equations. A plot of temperature distribution with state points is shown in Figure 5. This plot shows that the exhaust gas enters the waste-heat boiler at temperature T_{4b}, specified by the gas-turbine cycle. It exits at temperature T_{5b} which is specified by the mass flow of the working fluid for a given waste-heat boiler. The steam-cycle temperature entering the waste-heat boiler, T_b, is known from the baseline ship. The exit temperature of the steam is assumed to be 50^o below the entering gas temperature, T_4. This assumption is consistent with the waste-heat-boiler design from the baseline ship. A thermodynamic limit exists in the cycle as seen in Figure 5. The saturation temperature of the steam cannot be greater than the temperature of the gas, T_x, at the corresponding axial position. This temperature difference is known as the 'pinch' temperature. The First Law written for the heat transfer of the steam cycle from the start of the waste-heat boiler to this saturation temperature and

FIGURE 4

WASTE-HEAT STEAM CYCLE

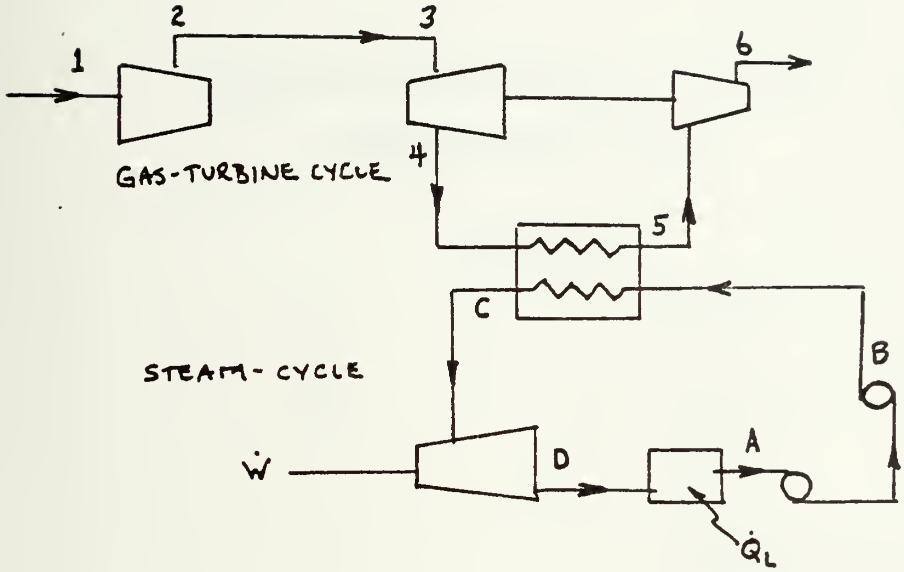


FIGURE 5

AXIAL-TEMPERATURE DISTRIBUTION

FOR A HEAT EXCHANGER

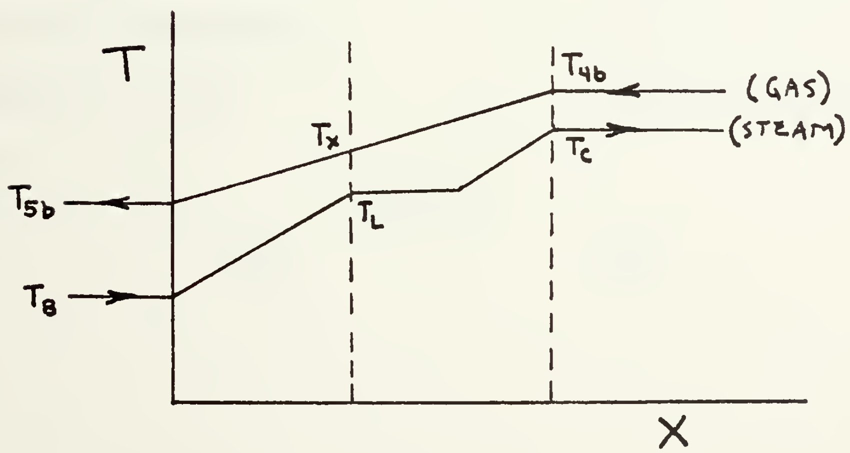


TABLE 3

STATES AND ASSUMPTIONS FOR THE STEAM-TURBINE CYCLE

STATES	
P_a	28" Hg
P_b	600 psi
P_c	600 psi
h_a	Saturated Liquid
h_b	69.75 BTU/lbm
h_c	Specified by T_c
GAS CYCLE	
Mass Flow of Gas	463,000 lbm/hr
Inlet Gas Temperature	Varies with Fan
Exit Gas Temperature	Varies with Fan
Pressure of Gas	Varies with Fan
ASSUME	
η_{is_T}	80%

again for the remainder of the waste-heat boiler results in the following equations:

$$\dot{m}_g(h_x - h_5) = \dot{m}_s (h_1 - h_b)$$

$$\dot{m}_g(h_4 - h_x) = \dot{m}_s (h_c - h_1)$$

There are two unknowns in the equations; $\dot{m}_s$, the mass flow of steam, and h_x, the enthalpy of the gas at the steam saturation point. These equations can be solved simultaneously to check the thermodynamic limit and find the mass flow of steam.

Referring again to Figure 4, the power out of the steam turbine is found from the following equation:

$$\dot{W} = \dot{m}_s (h_c - h_d)$$

The work into the steam cycle is found from the following equation:

$$\dot{W} = \dot{m}_s (h_b - h_a)$$

The net power from the steam cycle is the difference between the work out and the work into the cycle. This will change the total power available by combining the gas-turbine power with the steam power. Since there has been no additional outside heat source, the heat energy into the combined cycle is the same as the gas-turbine cycle. Thermal efficiency is the net work out of the combined cycles divided by the heat energy into the gas-turbine cycle.

2.3.5. INDUCED-DRAFT FAN

The objective of analyzing the induced-draft fan is to determine the number of stages, efficiency, size and speed of the fan. The first step to reach these objectives is to select a velocity diagram.

2.3.5.1. VELOCITY DIAGRAM

The velocity diagram is set by key assumptions which were made to reduce design risk while maintaining a high efficiency. Table 4 lists these assumptions along with the standard values for compressor design. The velocity diagram for this design is found in Figure 6. The resulting work coefficient (0.2369) is in the range for normally loaded axial machines with 2 or more stages. A summary of other angles and relations is found in Table 5.

2.3.5.2. NUMBER OF STAGES

The number of stages is a function of the total work and the work per stage. The work per stage is a function of the peripheral speed. The total work is a function of the total temperature as expressed in the following equation:

$$T_{02} = T_{01}\left(\frac{P_{02}}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}\cdot\frac{1}{\eta_p}}$$

TABLE 4

STATES AND ASSUMPTIONS FOR THE

INDUCED-DRAFT FAN

STATES

CONDITION	LM 2500	ALLISON
Inlet Pressure	Vary	Vary
Temperature at Inlet	Vary	Vary
Mass Flow	463,000 lbm/hr	109,370 lbm/hr
Exit Pressure	15.06 psia	15.06 psia

ASSUMPTIONS

γ	1.4

INITIAL VELOCITY DIAGRAM VALUES

PARAMETER	DESIGN PRACTICE	SELECTED
W_2/W_1	>0.72	0.80
ϕ	$0.35<\phi<0.8$	0.40
R_h/R_t	~ 0.7	0.75
		Axial Inlet Flow

FIGURE 6

VELOCITY DIAGRAM

FOR INDUCED-DRAFT FAN

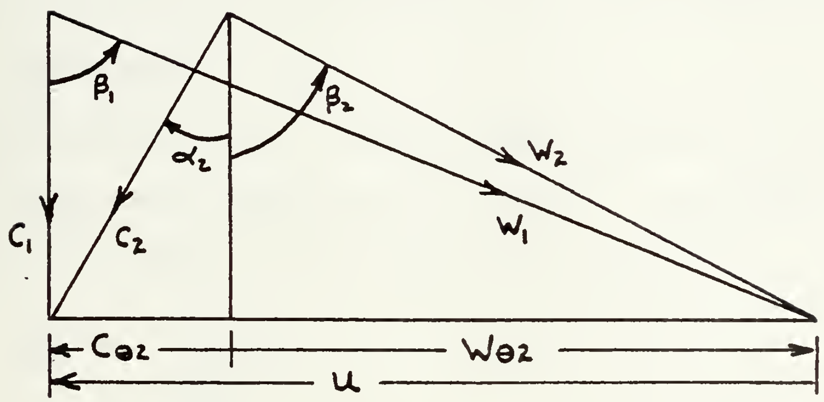


TABLE 5

SUMMARY OF

VELOCITY DIAGRAM

β_1	68.2°	α_1	0°
W_1	1.077u	C_1	0.4u
$W_{\theta 1}$	u	$C_{\theta 1}$	0
β_2	62.3°	α_2	30.6°
W_2	0.8616u	C_2	0.4649u
$W_{\theta 2}$	0.7631u	$C_{\theta 2}$	0.2369u

Since the efficiency is not known at this point, it must be assumed and the process iterated after the efficiency is calculated. The number of stages is specified, thus the work per stage and peripheral speed are established.

2.3.5.3. EFFICIENCY

The efficiency of the induced-draft fan can be calculated using basic equations for gases. Polytropic efficiency is defined by the following equation:

$$\eta_p \equiv \frac{\frac{\gamma - 1}{\gamma}}{\frac{n-1}{n}}$$

The assumed value for γ is 1.4. The value for n is defined by the process relation:

$$p\ v^n = \text{Constant}$$

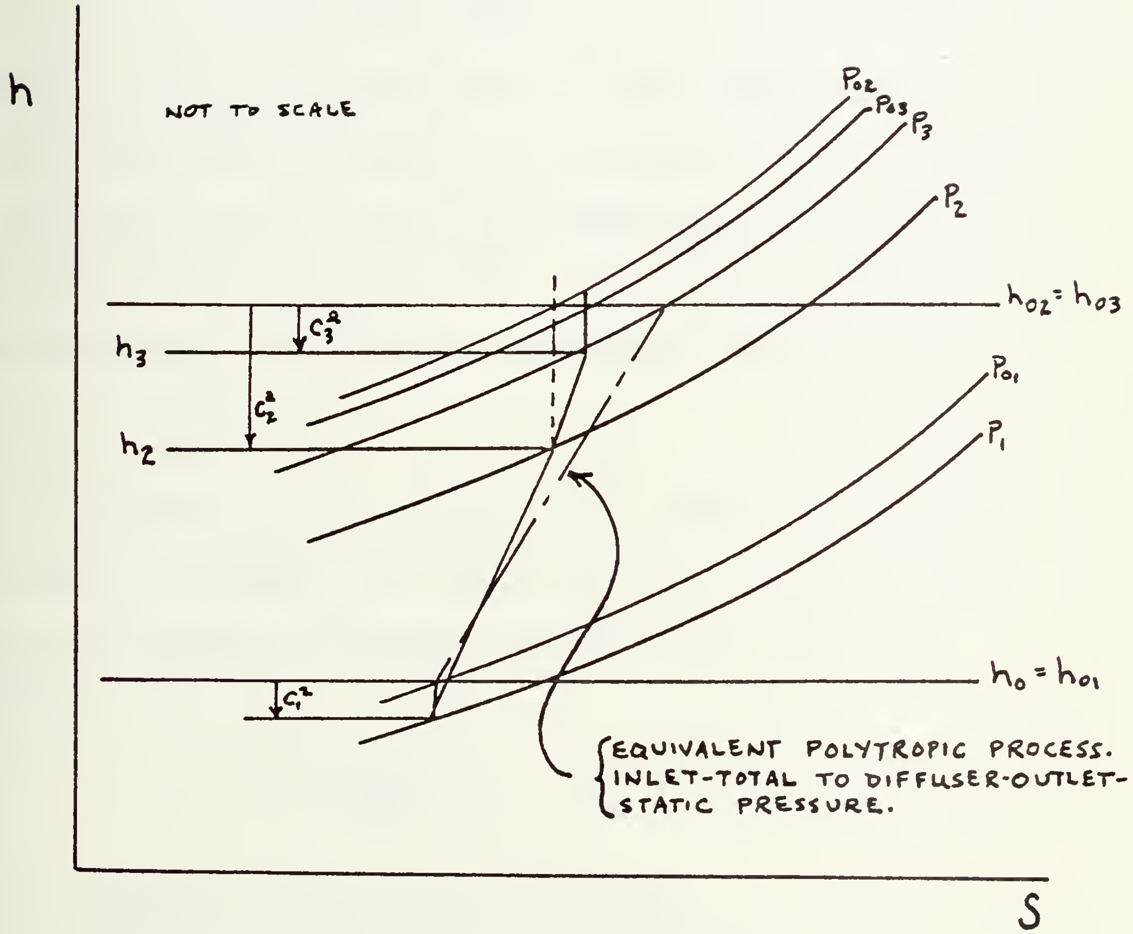
Since the total to static efficiency is desired, as shown in Figure 7, the static pressure must also be calculated to use in the gas relation. This can be found by calculating the Mach number which is a function of the exit velocity and total temperature which have already been found in previous sections.

The Mach number for the first stage is expected to be the highest and can be calculated from the velocity and total temperature also found in previous sections.

FIGURE 7

H-S DIAGRAM

FOR INDUCED-DRAFT FAN



2.3.5.4. SIZE

The objective of calculating the size of the machine is to determine the approximate diameter of the induced-draft fan. This can be accomplished by assuming a constant mid-radius through the machine and solving simultaneous equations for the conservation of mass, and area between concentric circles as expressed below.

$$\dot{m} = VA\rho$$

$$A_1 = (r_t^2 - r_h^2)\pi$$

The hub-tip ratio is specified for the first stage (0.75). As shown in Table 5, the velocity is a function of peripheral speed, which was established when the number of stages was specified. The density can be calculated from the pressure and temperature.

2.3.5.5. SPEED

Peripheral speed, u, was established by specifying the number of stages. The RPM of the induced-draft fan is found from the peripheral speed by the following equation:

$$N = \frac{30\ u}{r_m \pi}$$

2.3.6. SUMMARY OF ANALYTICAL APPROACH

Calculations for the gas-turbine cycles, steam cycle, and induced-draft fan have been reduced to First Law

equations, Equations of State, Conservation of Mass and other simple relations. The several assumptions which have been made have been consistent with good design practice and have not been overly optimistic. The design of the induced-draft fan emphasized low design risk while maintaining adequate efficiency. The validity of this approach will be shown by a comparison of results with actual design ratings in later sections. The detailed calculations for the gas-turbine cycles, the steam cycle, and the induced-draft fan are found in Appendices A, B, and C respectively.

2.4. RESULTS USING THE CYCLE FOR GENERATION OF ELECTRICAL POWER

This section will present the power and efficiency results of the inverted Brayton cycle for the electrical-power-generation system. These results will be compared with the baseline Allison gas turbine with waste-heat boiler. Additionally, the gas-turbine calculations without the waste-heat boiler are included to demonstrate the validity of the results.

2.4.1. BRAYTON CYCLE

The calculations for the gas-turbine cycle described in section 2.3.1 were completed for the Allison gas turbine without the waste-heat boiler or induced-draft fan. The gas-turbine power was calculated as 3,426.9 hp.

This compares with the rated power of 3,640 hp for the same climatic condition as listed in Table 1. The methodology can be considered to give valid results for this gas-turbine cycle.

2.4.2. EFFECT ON POWER OF ADDING A WASTE-HEAT BOILER

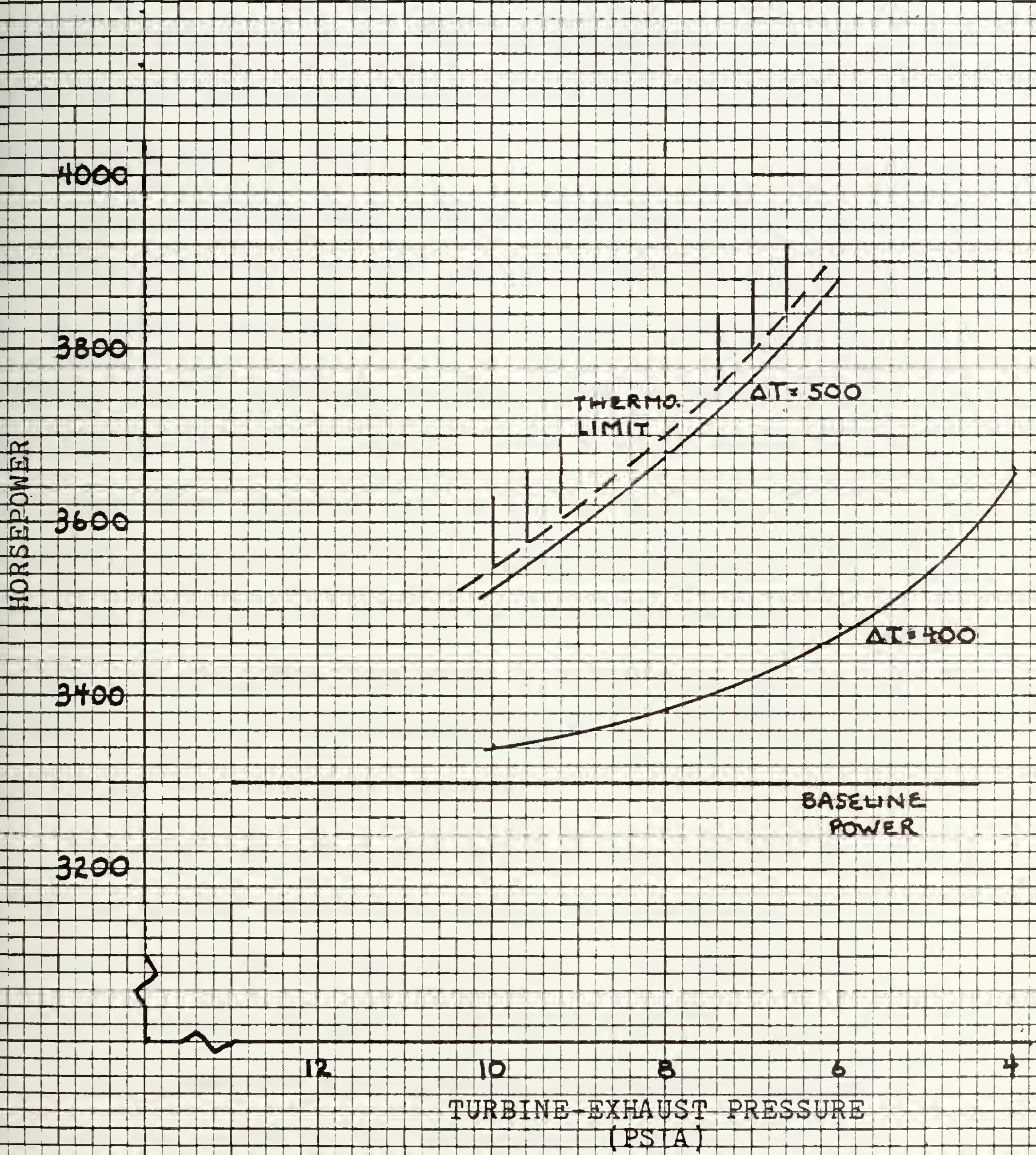
The addition of the waste-heat boiler to the Allison gas turbine should decrease the power from the gas turbine as described in section 2.3.2 because of the backpressure. The result of the calculations was a 4% loss in gas-turbine power, to a new rating of 3,299.8 hp. This will be used as the baseline power rating of the Allison gas-turbine for the remainder of the study.

2.4.3. EFFECT ON POWER OF ADDING AN INDUCED-DRAFT FAN

The effect of adding an induced-draft fan to the Allison gas turbine was investigated using the analysis method presented in section 2.3.3. The pressure at the exit flange of the gas turbine was varied in increments of 2 psia to a lower limit of 6 psia. The baseline DD 963 waste-heat boiler was used as well as another smaller boiler measured in size by the change in temperature across the gas side of the heat exchanger. A plot of the results of varying the pressure ratio of the induced-draft fan and waste-heat boiler is shown in Figure 8. Absolute pressure at the exhaust flange of

FIGURE 8

EFFECT OF INDUCED-DRAFT FAN
ON ALLISON GAS-TURBINE POWER



the gas turbine is measured against the net horsepower of the gas turbine. The two waste-heat boilers are indicated by lines of constant ΔT. The baseline power for the gas-turbine with waste-heat boiler is indicated by the horizontal line. The thermodynamic limit in this case is the saturation temperature of the steam and is shown as a dashed line.

The resulting gas-turbine power is a function of the pressure ratio of the induced-draft fan and the size of the waste-heat boiler. For a given boiler, gas-turbine power increases as the pressure ratio of the induced-draft fan increases. For a given induced-draft fan, gas-turbine power increases as the size of the waste-heat boiler increases. These results are consistent with the explanation of the inverted Brayton cycle; a cooler gas requires less work to compress, and a larger pressure ratio in the induced-draft fan increases the net power from the gas turbine.

Gas-turbine power increases from 6 to 12% can be obtained as the saturation limit is approached if the induced-draft fan is satisfactory. The baseline waste-heat boiler was evidently designed close to this limit.

The effect on the steam cycle is not calculated because the steam is not converted to power for this application. The steam is used for heat energy with

excess consumed by a control condenser. Although there would be an expected decrease in this heat energy available as a result of adding the induced-draft fan, the decrease is not part of the power cycle.

2.4.4. EFFECT ON EFFICIENCY OF ADDING AN INDUCED DRAFT FAN

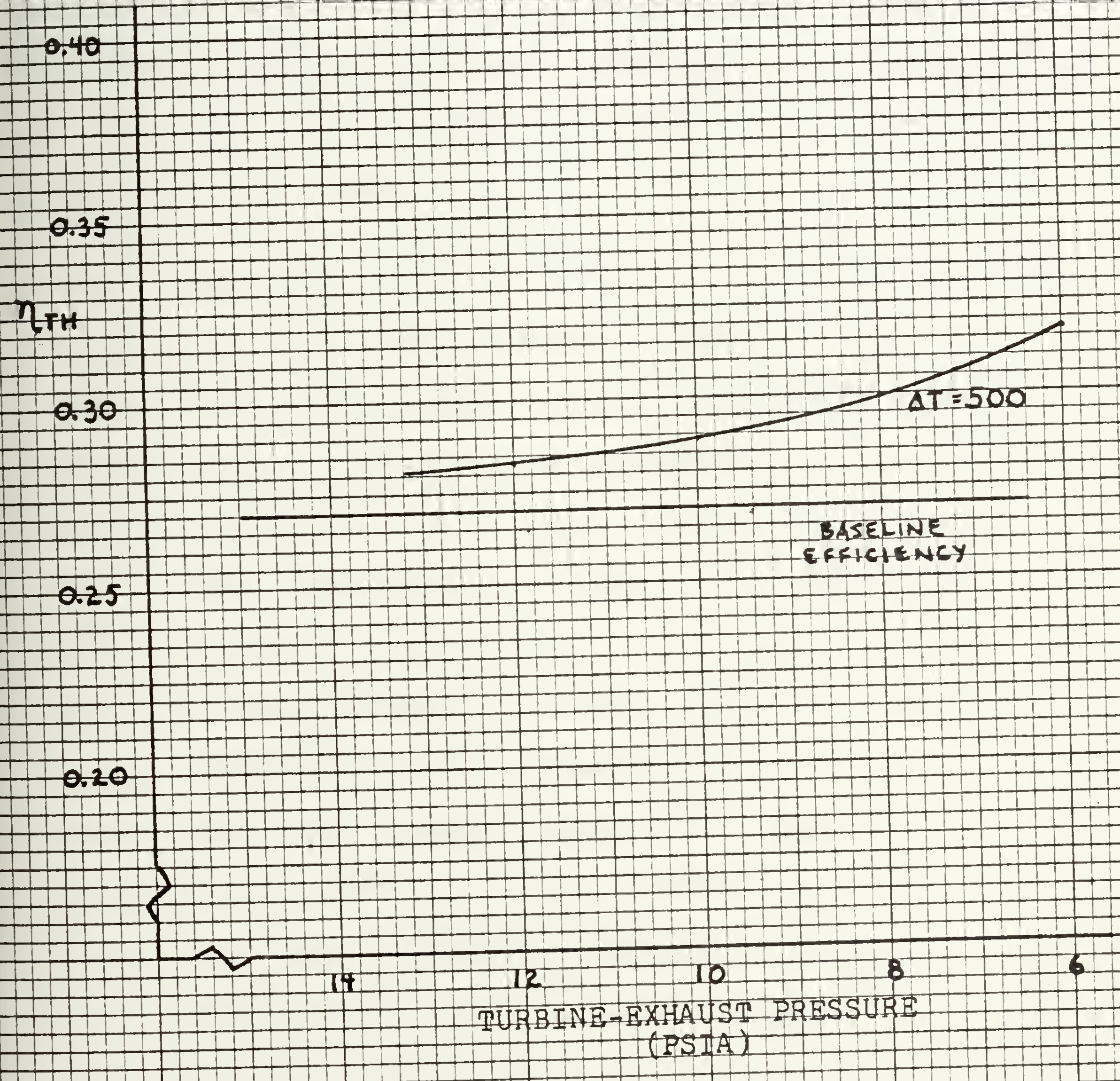
Thermal efficiency of the inverted Brayton cycle was calculated using the approach described in section 2.3.3. Figure 9 displays the results of those calculations with regards to the Allison gas turbine with the waste-heat boiler but without the induced-draft fan. Increasing the pressure ratio of the induced-draft fan is again indicated by the decreasing value of the absolute pressure at the exhaust flange of the gas turbine. The results are consistent with the power findings in that cycle efficiency increases as the size of the induced-draft fan increases up to the thermodynamic limit of saturated steam.

2.4.5. DESIGN OF THE INDUCED-DRAFT FAN

An induced-draft fan was designed for the Allison gas turbine using the approach outlined in section 2.3.5. An attempt to achieve the 12% power gain with less than four stages in the fan with a specified maximum diameter resulted unsatisfactorily in supersonic flow in the fan. A satisfactory design was achieved for a 9% power gain.

FIGURE 9

EFFECT OF INDUCED-DRAFT FAN
ON ALLISON GAS-TURBINE EFFICIENCY



The efficiency of this induced-draft fan peaks at three stages as seen in Figure 10. A summary of the desired design characteristics is shown in Table 6. This design requires at least 3 stages to have subsonic flow. The remaining criteria of RPM and diameter are satisfactory in the three-stage design. The exhaust ducting in the gas turbine has a five-foot diameter, and the three-stage fan design would be less than three feet. An induced-draft fan with low risk and good performance in its design is possible for the desired power.

2.4.6. CONCLUSIONS ON THE GENERATION OF ELECTRICAL POWER

The baseline electrical-power-generation cycle consists of an Allison gas turbine with a DD 963 waste-heat boiler and a total rated horsepower of 3299.8 hp. Various pressure ratios for induced-draft fans were added to this cycle with the effect of increasing the gas-turbine power significantly. The baseline waste-heat boiler represented an optimum design with regards to the thermodynamic limit of saturated steam. An induced-draft fan with low risk and good performance is possible for a 9% gain in gas turbine power and efficiency. These significant gains are worthy of further investigation for cost-effectiveness.

FIGURE 10

EFFICIENCY DIAGRAM

FOR ALLISON INDUCED-DRAFT FAN

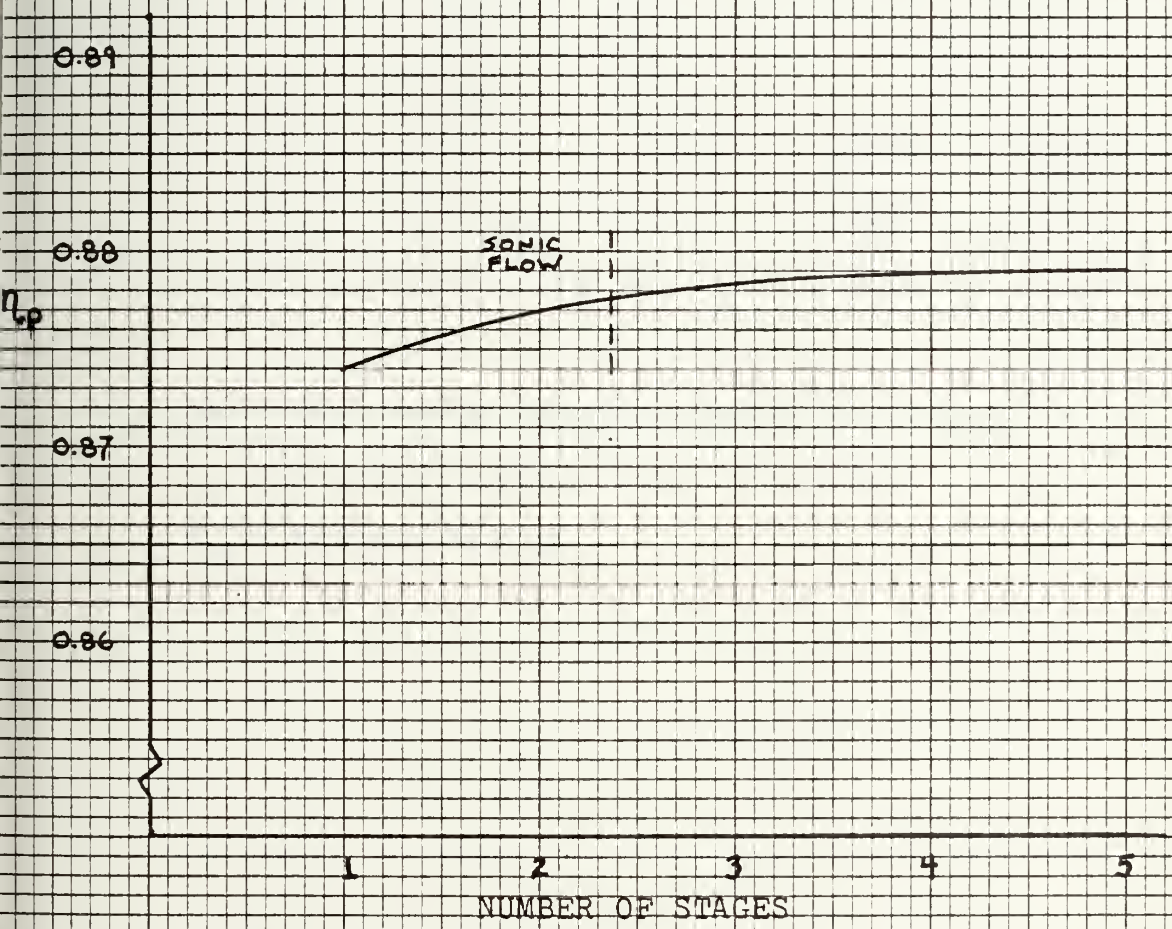


TABLE 6

SUMMARY OF INDUCED-DRAFT FAN
FOR ALLISON GAS TURBINE

# STAGES	M_1	η_p	RPM	R_{t1} (ft)
1	1.838	0.874		
2	1.015	0.877		
3	0.881	0.878	11230	1.328
4	0.749	0.879	9048	1.427
5	0.662	0.879	7654	1.509

M_1 = Mach Number at First-Stage Rotor Tip

η_p = Compressor Polytropic Efficiency

R_{t1} = Maximum Radius

2.5. PROPULSION CYCLE

This section will present the power and efficiency of the inverted Brayton cycle for the propulsion cycle. These results will be compared with the baseline propulsion plant consisting of the LM2500 gas turbine with waste-heat boiler to demonstrate that no additional power is gained with this cycle. The ship designer might be misled by the 9 percent power gain obtained by adding an induced-draft fan to the gas-turbine generator in the electrical cycle. This gain should not be expected for the COGAS propulsion system because in this cycle the steam is put back into propulsive power. The gas turbine calculations without the waste-heat boiler are included to demonstrate the validity of the results.

2.5.1. BRAYTON CYCLE

The gas turbine cycle calculations described in section 2.3.1 were completed for the LM2500 propulsion gas turbine without the waste heat boiler or induced draft fan. The gas turbine power was calculated as 21,459.6 hp. This compares with the rated power of 21,500 hp for the same climatic condition as listed in Table 1. The methodology can be considered to give valid results for the gas turbine cycle.

2.5.2. EFFECT ON POWER OF ADDING A WASTE-HEAT BOILER

The addition of the waste heat boiler to the propulsion gas turbine should decrease the power from the gas turbine as described in section 2.3.2 because of the backpressure. The result of the calculations was a 3% loss in gas-turbine power, to a new rating of 20,865 hp.

Steam-cycle power was calculated using the method described in section 2.3.4. An additional 7795 hp was provided by the steam cycle. Again, this result compares closely with the value found by the DG/AEGIS study [7] and also by Abbott [2] . This increase in horsepower represents a 36% gain over the original gas turbine and will be used for a baseline combined-plant power rating.

2.5.3. EFFECT ON POWER OF ADDING AN INDUCED DRAFT FAN

The induced-draft fan is added to the baseline propulsion system of the LM2500 gas turbine with the waste-heat boiler and the steam turbine to demonstrate the failure to gain power in this cycle. The effects of the induced-draft fan will be investigated from three views; the effect on the gas-turbine power, the effect on the steam-turbine power, and the effect on combining steam and gas-turbine power.

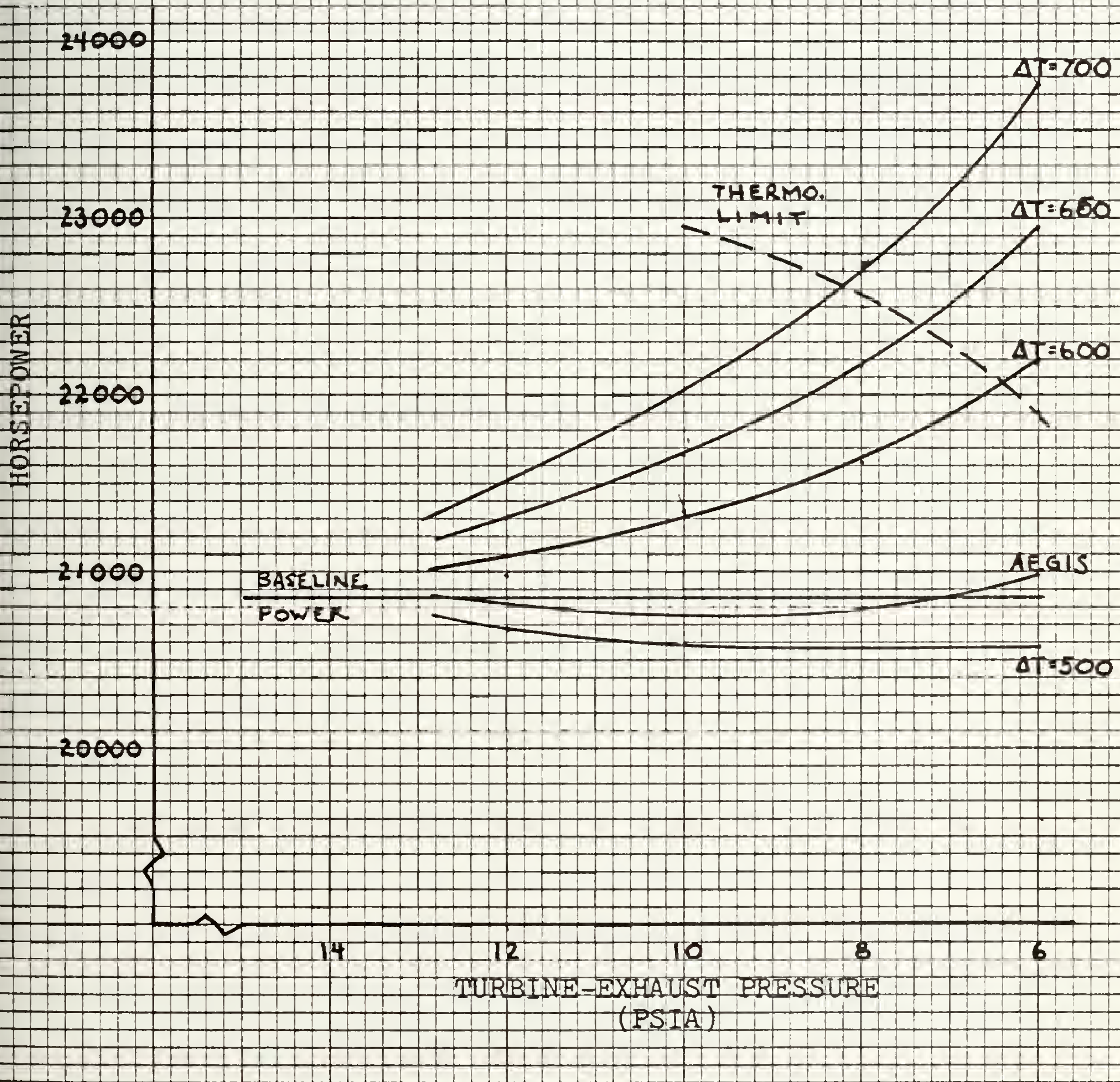
2.5.3.1. EFFECT ON GAS-TURBINE POWER

The effect of adding an induced-draft fan to the gas-turbine power was investigated using the analysis method presented in section 2.3.3. The pressure at the exit flange of the gas turbine was varied in increments of 2 psia to a lower limit of 6 psia. The baseline DG/AEGIS waste-heat boiler was used as well as other potential boilers measured in size by the change in temperature (ΔT) across the gas side of the heat exchanger. A plot of the results of varying the pressure ratio of the induced-draft fan and waste-heat boiler is shown in Figure 11. Absolute pressure at exhaust flange of the gas-turbine is measured against the net horsepower of the gas turbine. Several sizes of waste-heat boilers are indicated, including the baseline, by lines of constant ΔT. The baseline power of the gas turbine with waste-heat boiler is indicated by the horizontal line. The thermodynamic limit posed by the 'pinch' temperature, as discussed in section 2.3.4, is shown as a dashed line.

The resulting gas-turbine power is a function of the pressure ratio of the induced-draft fan and the size of the waste-heat boiler. For a given boiler, gas-turbine power increases as the pressure ratio of the induced-draft fan increases. For a given induced-draft fan, gas-turbine power increases as the size of the waste-heat boiler

FIGURE 11

EFFECT OF INDUCED-DRAFT FAN
ON LM2500 GAS-TURBINE POWER



increases. These relationships are consistent with the results from the Allison gas turbine in the electrical cycle.

The addition of an induced draft fan might directly result in a decrease of gas-turbine power as seen by the plot for the smallest-sized waste-heat boiler. The pressure losses in the system were sufficient to overcome the gain from the fan in this case.

Gas-turbine power increases of 6-7% are obtained for the larger waste-heat boilers as the thermodynamic limit is approached. The baseline DG/AEGIS waste-heat boiler shows a small gain in power for the larger induced-draft fans.

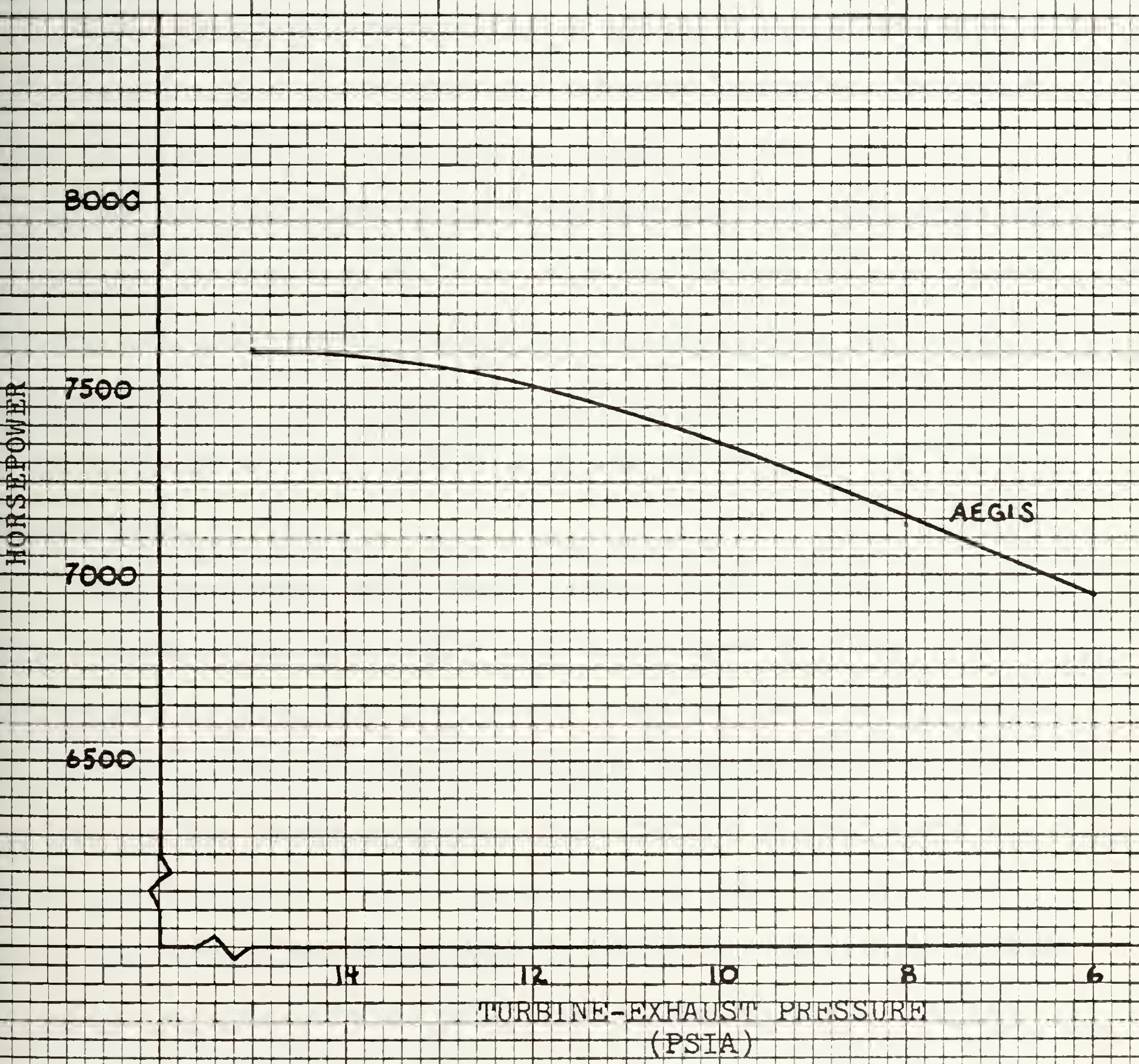
2.5.3.2. EFFECT ON STEAM-TURBINE POWER

The steam-cycle power was calculated using the method described in section 2.3.4 and the results plotted in Figure 12 for the baseline DG/AEGIS waste-heat boiler. The power for the steam cycle is plotted against the absolute pressure at the exhaust flange of the gas turbine.

The plot shows steam-turbine power decreasing as the pressure ratio of the induced-draft fan is increased. This effect was anticipated in section 2.3.4. The reason the steam-turbine power decreases is because increasing the pressure ratio of the induced-draft fan decreases the

FIGURE 12

EFFECT OF INDUCED-DRAFT FAN
ON DG/AEGIS STEAM-TURBINE POWER



absolute pressure at the exhaust flange of the gas turbine with an accompanying decrease in temperature. This lower temperature of the gas-turbine exhaust decreases the quantity of heat energy available to the steam in the waste-heat boiler.

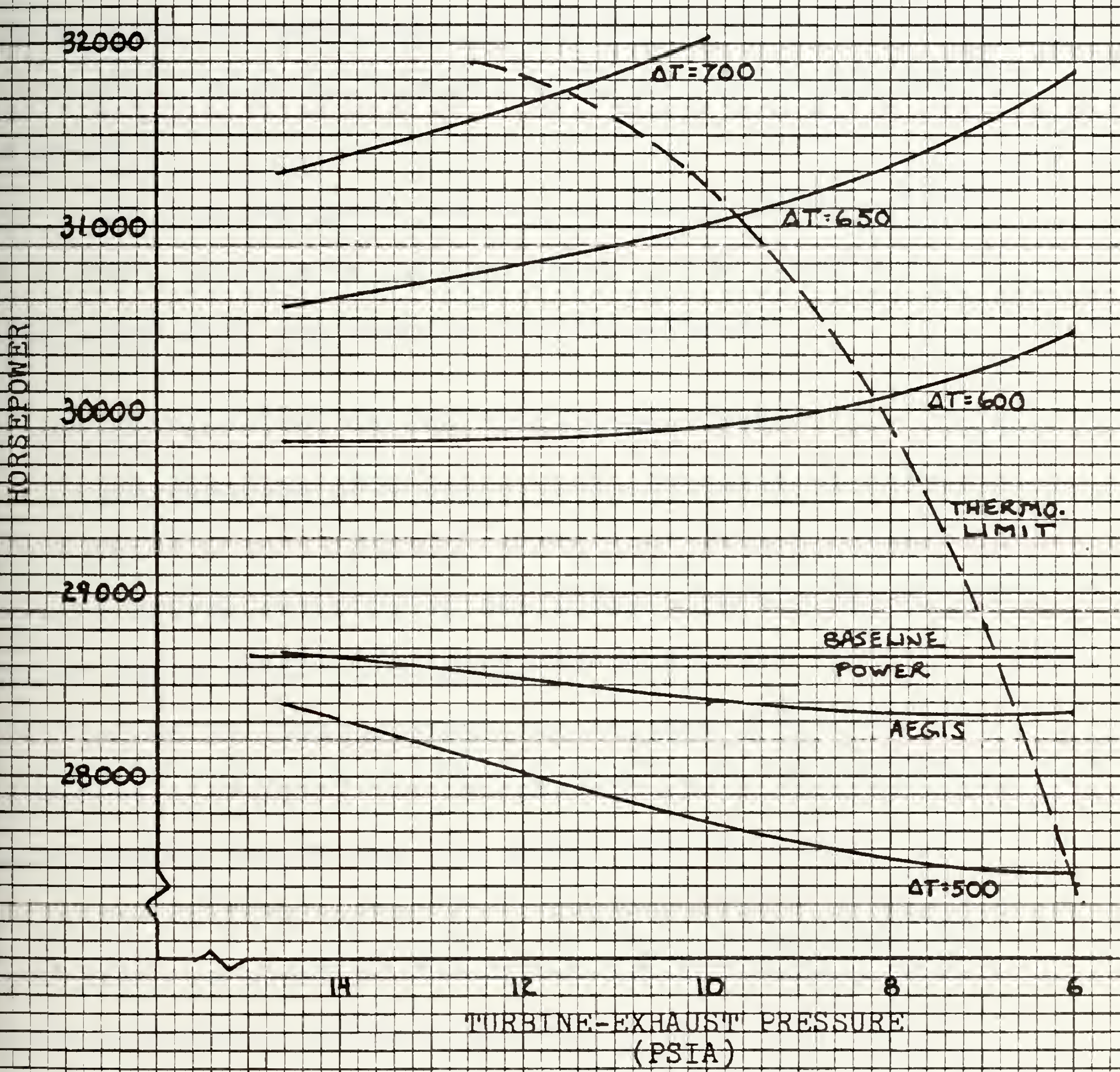
2.5.3.3. EFFECT ON THE COMBINED CYCLES

The result of combining the gas-turbine power and the steam-turbine power for the inverted Brayton cycle is shown in Figure 13. Absolute pressure at the exhaust flange of the gas turbine is measured against the combined horsepower of the two cycles. Several sizes of waste-heat boilers are indicated, including the baseline, by lines of constant ΔT. The baseline combined-cycle horsepower is indicated as well as the thermodynamic limit of 'pinch' temperature.

For the baseline DG/AEGIS waste-heat boiler, the gain from the induced-draft fan in the gas turbine is not sufficient to overcome the power loss in the steam turbine. Larger waste heat boilers show some improvement but are cut off by the thermodynamic limit before significant gains are achieved. This demonstrates that the induced-draft fan offers no advantage in power for this application of combining gas-turbine and steam-turbine power for the propulsion system.

FIGURE 13

EFFECT OF INDUCED-DRAFT FAN
ON LM2500 GAS-TURBINE AND
STEAM-TURBINE POWER COMBINED



2.5.4. EFFECT ON EFFICIENCY OF ADDING AN INDUCED-DRAFT FAN

Thermal efficiency of the inverted Brayton cycle was calculated using the approaches described in sections 2.3.3 and 2.3.4 for the cycles with and without the steam-turbine power. Figure 14 displays the results of those calculations with regard to the baseline propulsion system with the waste-heat boiler but without the induced-draft fan. Only the baseline DG/AEGIS waste-heat boiler is shown.

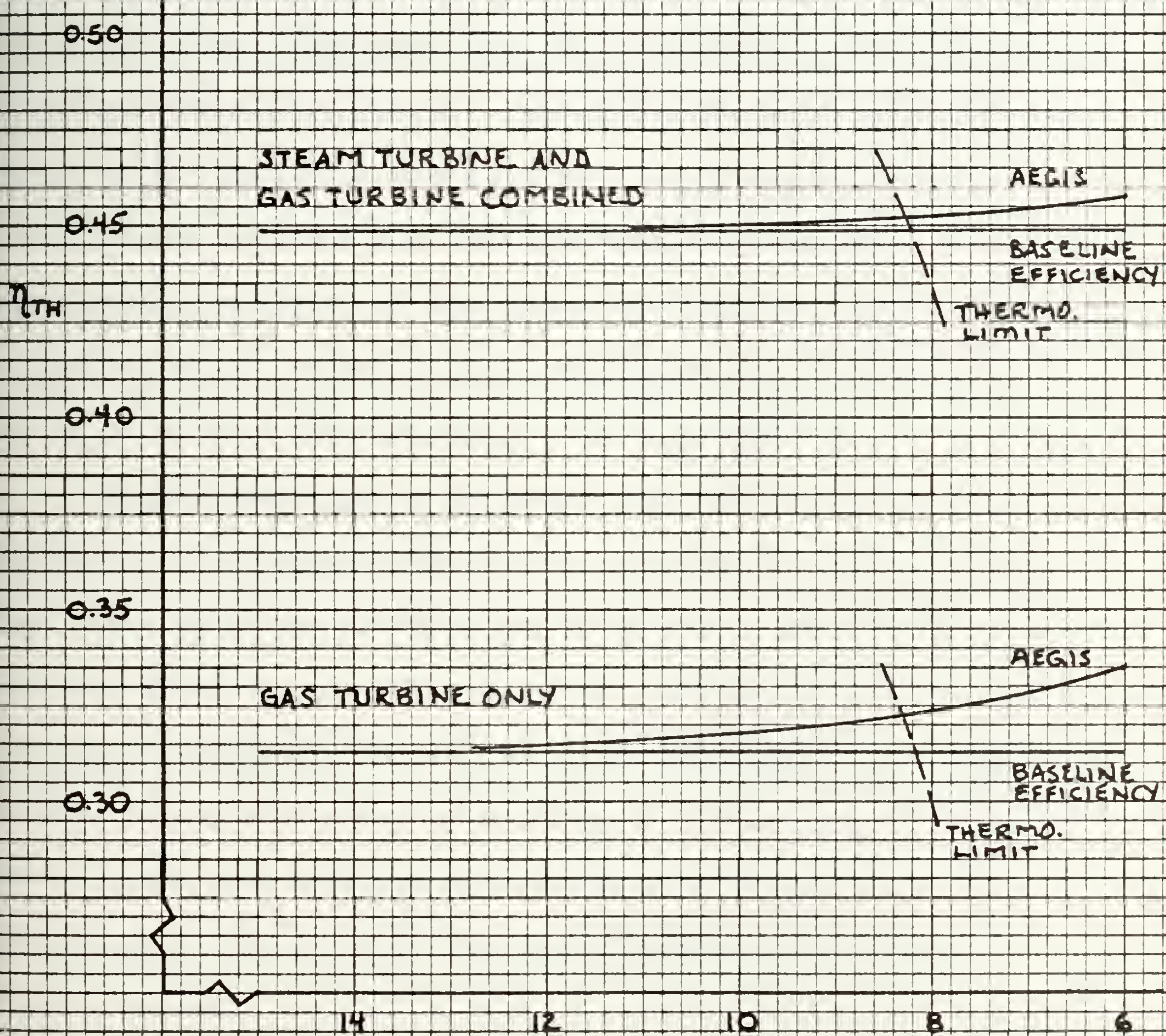
The induced-draft fan has an insignificant effect on thermal efficiency for the cycle. This result is consistent with the results for cycle power.

2.5.5. CONCLUSIONS ON THE PROPULSION CYCLE

The baseline propulsion cycle consists of an LM2500 gas turbine with a DG/AEGIS waste-heat boiler and a steam turbine driven entirely by the steam from the waste-heat boiler for a total-rated power of 28,660 hp. The addition of an induced-draft fan to this cycle has the effect of increasing the gas-turbine power slightly but concurrently decreasing the steam-turbine power. As anticipated, power gains from the gas turbine are not sufficient to overcome the losses in the steam turbine. Cycle efficiency reflects similar results with insignificant gains inside the thermodynamic limit. Although it can be said the calculations corresponded accurately

FIGURE 14

EFFECT OF INDUCED-DRAFT FAN
ON LM2500 GAS-TURBINE EFFICIENCY



TURBINE-EXHAUST PRESSURE
(PSIA)

with published rated horsepowers, indicating a valid approach, the application of the inverted Brayton cycle for the baseline propulsion system or any other COGAS cycle will show no worthwhile gain in horsepower or efficiency.

2.6. CONCLUSIONS FROM THE THERMODYNAMIC ANALYSIS

The power and efficiency of the inverted Brayton cycle have been analyzed for the propulsion and electrical-power generation systems. The thermodynamic approach was shown to be valid by the accuracy in predicting the rated horsepower of the two gas turbines studied. The gas-turbine portion of the inverted Brayton cycle performed as anticipated, namely, gas-turbine power increased as the size of the induced-draft fan increased.

Significant gains in power and efficiency were achieved in the electrical-power-generation application. The steam produced in the baseline waste-heat boiler was not converted to electrical power through a turbine but was used elsewhere as heat energy. The 9% gain in gas-turbine power achieved by adding the induced draft fan to the Allison gas turbine indicates this application of the Brayton cycle should be investigated for cost-effectiveness.

The promising gains from the electrical cycle should not be misinterpreted by the ship designer as representative for adding induced-draft fans to any gas-turbine cycle.

For a COGAS propulsion system the decrease in steam-turbine power caused by adding the induced-draft fan will offset the gains from the gas-turbine power. It is important to point out that the COGAS system does increase propulsion power by 30% over the gas turbine alone for the LM2500. But the inverted Brayton cycle will not improve on this gain. Thermodynamically, there is no purpose in applying the inverted Brayton cycle to a COGAS propulsion system.

CHAPTER 3 CYCLE UTILIZATION

3.1. INTRODUCTION

The results of the thermodynamic analysis have shown that gains in horsepower and efficiency are possible with the proper application of an inverted Brayton cycle. The objective of this chapter is to perform a complete ship-systems analysis to determine if the advantage in horsepower and efficiency can be utilized on a naval combatant. This goal will be achieved by determining if there is an increase in cost-effectiveness in the application.

A method of classifying the options available for utilizing prospective gains in power and efficiency will be presented. One option includes increasing the performance of the baseline ship with an accompanying 'cost' penalty. Another option would be to maintain a constant performance and decrease the 'cost'. Options for either propulsion or electrical-power generation will be outlined to demonstrate the flexibility of the approach.

Using this method, the measurement of performance and 'cost' will be accomplished to determine the cost-effectiveness for any option. In defining the factors involved in performance and 'cost', the ship-system

approach will be utilized. There will be no attempt to give pre-engineering judgement as to the significance of any of the factors so that the approach will be investigated completely. Since the propulsion application of the inverted Brayton cycle is known to be not worthwhile, only the options available to the Allison gas-turbine application will be computed.

3.2. OPTIONS FOR CYCLE UTILIZATION

The goal of the naval ship designer is to maximize the cost-effectiveness of the ship. Currently, this is being done by maximizing performance while holding cost as a constant. This procedure is known as "design to cost". Another alternative for maximizing cost-effectiveness would be to fix performance at a constant and minimize the cost. Either option depends on investigating all of the design elements in the ship to measure their performance and cost.

The various design elements are interdependent on each other. Increasing the performance of a subsystem would normally be accomplished with an increase in the component weight, volume, or manning. The direct weight increase of the subsystem will cause a weight increase in several other subsystems in order to maintain a constant level of performance for the ship. Graham [10] reports

that it is not unusual for the ratio of direct to total-ship-impact weight to be on the order of 1 to 10 for certain complex subsystems. In his example, a sonar subsystem had a direct weight of only 60 tons, but resulted in a total-ship impact in terms of the increase in ship displacement of 600 tons. This type of increase would most probably be in conflict with the ship design philosophy. The point is that the subsystem designer should approach his design task with the idea of optimizing the overall ship system and not suboptimizing the ship system while attempting to optimize his particular subsystem.

When evaluating alternative approaches to a subsystem design, the conclusion should be based on the "true cost" of the subsystem. True cost implies not only consideration of the direct but also the indirect costs involved in incorporating the subsystem into the ship. The direct costs include the machinery, fuel and manning of the subsystem. The indirect costs include ship growth as a result of the added machinery in terms of weight, volume, electrical power, fuel and manning in the rest of the ship.

3.2.1. FACTORS CONSIDERED IN ANALYZING NAVAL SHIP SYSTEMS

The 'true cost' of the subsystem is only one of two factors needed to measure the cost-effectiveness of sub-

system alternatives. Performance is the other factor. The following sections will identify and define the elements of these factors used to analyze naval ship systems.

3.2.1.1. TRUE COST

True cost has previously been described as the direct and indirect costs involved in incorporating the subsystem into the ship. A variation in the organization of the elements of true cost is useful when one is concerned with total life-cycle cost. The elements can be divided into the two categories shown in Figure 15, acquisition cost, and operating cost.

The first element shown under acquisition cost is machinery cost. This would include the acquisition and installation cost of the subsystem machinery. Test and development costs can be considered as part of the acquisition cost.

Ship impact cost is the other element of acquisition cost. The influence of a design feature is felt through its demand of four parameters which are basic to design; weight, space, manning and electrical power. This influence is applied to the total ship and reflects the ship growth which accompanies subsystem growth.

Run cost is the first element listed under operating cost. It can be divided into direct run cost of the

FIGURE 15

FACTORS IN MEASURING
TOTAL SHIP-SYSTEM COST

TOTAL LIFE-CYCLE COSTS

ACQUISITION COSTS	OPERATING COSTS
1. MACHINERY COSTS	1. RUN COST (FUEL)
ACQUISITION	2. MANNING
INSTALLATION	3. MAINTENANCE
2. SHIP IMPACT COSTS	

subsystem and indirect run cost of the ship. The direct run cost is the fuel consumed at the operation point of the subsystem. Indirect run cost is the greater electrical-power generation or propulsion fuel required to maintain constant performance.

Manning cost is also divided into direct and indirect sub-elements. The direct manning cost is the number of men required to operate and control the subsystem. Indirect manning cost is incurred because an increase in a sub-system level of manning has an iterative effect on the manning required to support the ship.

The final element of operating cost is maintenance. The cost of the men to maintain each subsystem, when different from the operating and control personnel, is included here. Additionally, repair parts and supply personnel to monitor spares and outside maintenance support are included.

3.2.1.2 PERFORMANCE FACTORS

The other factor necessary to analyze naval ship systems is performance. Figure 16 outlines the elements of performance by dividing them into two areas, systems-engineering performance and operational performance. The operational elements are the direct areas of basic performance. For a propulsion system, they would include speed, range, maneuverability and perhaps electrical power.

FIGURE 16

FACTORS IN MEASURING
TOTAL SHIP-SYSTEM PERFORMANCE

PERFORMANCE

SYSTEMS-ENGINEERING PERFORMANCE	BASIC OPERATIONAL PERFORMANCE
1. FLEXIBILITY	1. SPEED
2. R,M & A	2. ENDURANCE (RANGE)
3. STANDARDIZATION	3. ELECTRICAL POWER
4. RISK	4. MANEUVERABILITY AND RESPONSE
5. SURVIVABILITY	
6. SIGNATURE	

Systems-engineering elements are the indirect areas of performance. Again, for a propulsion system, the elements would include flexibility, standardization, risk, R,M&A, survivability and signature. There would be other elements for other systems, but the ones in Figure 16 represent the elements for this study.

Flexibility, the first element to be defined, is the ability to meet system functional requirements with a variety of configuration operating modes. In particular, this refers to the amount and the type of operational redundancy provided, not necessarily by identical backup units but also by alternate functional sources. The importance of achieving system flexibility is that it may help reduce either the number or size of individual hardware components needed. By this means, improvements can be realized in the other categories (cost, weight, etc.) without sacrificing the availability or the operational capability on the ship-system level.

Availability is the measure of the degree to which a system (or a component) is able to start performing its function at the start of any random mission. Reliability is the probability that the system will successfully perform its function without interruption once undertaken. Maintainability is the probability that an item will be restored to service within a given

period of time when prescribed maintenance is performed. The well known terms of mean time between failure (MTBF) and mean time to repair (MTTR) then determine the availability of the system (component) as follows:

$$A = \frac{MTBF}{MTBF + MTTR}$$

Failure is defined as an occurrence which causes the system (component) to become inoperative or perform in a degraded mode. Increasing MTBF to increase availability could thus be achieved either through redundancy of hardware or through more reliable components.

The definition of standardization here is the use of identical, interchangeable, pieces of equipment for any given function, i.e., all fired boilers for steam, all motor-driven compressors for air, etc. Standardization is desirable from the standpoint of both logistic support and maintenance training.

Technical risk is the measure of unproven capability that exists for a proposed design (component or system) in achieving the physical or functional performance expected.

Survivability, as applicable to military ships, is defined as the probability of losing mission capability as a result of enemy fire through the loss of electric power, propulsion, ship control or direct damage. The definition includes a series of hypothetical damage

situations such as flooding 15 percent of the ship at the waterline, stability under flooding conditions, etc. With regard to machinery systems, survivability translates into requirements for certain capabilities with one or more components of the system inoperative.

The final element in systems-engineering performance is signature. Signature is composed of the noise and temperature of the system as measured by the appropriate method. Design standards are usually set for critical signature sources. Warships desire low levels of noise from the ship to reduce the range of an enemy submarine's detection and classification of the ship. Low exhaust-stack temperatures are desirable to minimize the target size of incoming infrared-seeking missles.

The first three operational performance factors studied in this report concern the propulsion system. Speed is the maximum velocity the ship can attain for given climatic and hull conditions. The endurance is the range the ship can travel starting with full fuel tanks, transiting at an optimum speed. This speed is set prior to the ship design, and the propulsion system should reflect optimum efficiency at this speed. Maneuverability and response reflect the handling characteristics of the propulsion system.

The final performance factor is the electrical-power capacity. Standard practice in naval ship design

calls for the battle load to be carried with at least one generator not in use. Additionally, there is a margin for growth which should not be encroached during the design stage.

3.2.1.3. SUMMARY OF FACTORS CONSIDERED

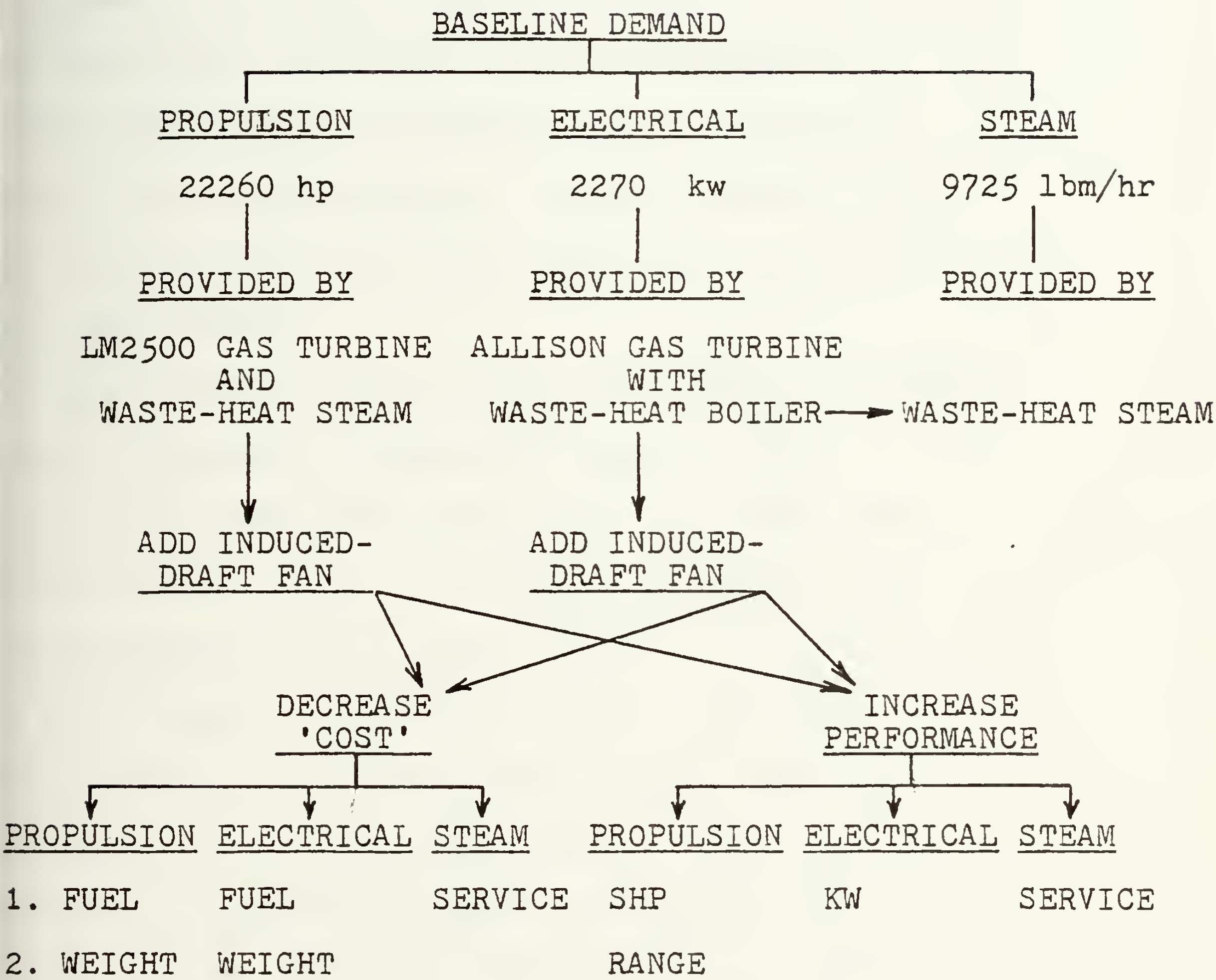
The factors considered in analyzing naval ship systems were divided into two categories, cost factors and performance factors. The cost factors were analyzed from a life-cycle cost structure with elements including acquisition cost and operating cost. Each subsystem cost and its iterative impact on the total ship were described. The performance factors included systems-engineering performance and operational performance. A later section will measure the cost of the factors for the application with the inverted Brayton cycle, but the factors apply to any subsystem optimization.

3.2.2. COST/PERFORMANCE OPTIONS

The power and efficiency gains obtained from the application of the inverted Brayton cycle to the electrical-power-generation system can be utilized in several different ways. Figure 17 depicts these options as well as possibilities from future gains in propulsion. The baseline energy demands for the electrical and auxiliary systems are included as described by Abbott [1].

FIGURE 17

COST/PERFORMANCE OPTIONS



gram shows a relationship between the electrical pulsion systems which is intended to demonstrate al-ship impact of any subsystem.

e branch of the option tree found in Figure 17 erformance being held nearly constant. With ed efficiency available, the fuel consumption drop, resulting in decreased cost and potential es in cost-effectiveness. This will occur if the of the increased efficiency does not carry too f a ship impact.

he other branch of the option tree shows performance sing as a result of increased power or fuel bility from decreased consumption. In this case, st to the ship system is certain to increase and st-effectiveness may either increase or decrease.

or either option, the impact can be transferred to sive, electrical, or auxiliary systems from the of the power and efficiency gains. For instance, savings in electrical-power generation can be increase the endurance range of the propulsive for an increase in performance, but with increased esulting from the induced-draft fan.

MEASURING COST-EFFECTIVENESS

he power and efficiency gains from the application

of the inverted Brayton cycle to the electrical-power-generation system will be applied to the option tree of Figure 17 and the cost measured by the 'true cost' factors defined previously. An increase in cost-effectiveness indicates a system with a positive impact on the total-ship system. Only the change in the cost to the ship will be estimated, not the absolute ship cost.

3.3.1. MACHINERY ACQUISITION COST

The only machinery added to the baseline Allison gas-turbine system was the induced-draft fan. The method proposed by Dunteman [6] is suitable for estimating the acquisition cost of the induced-draft fan. This method consists of estimating a base cost for an average-size and average-duty component and then varying the cost with size and duty. Cost factors for pressure ratio, hub-tip ratio, vacuum, number of stages, and RPM as well as base cost for the component are found in Figures 18 through 23. The input data from the induced-draft fan for the Allison gas turbine is shown in Table 7 as well as the results for the machinery cost. This cost was corrected to 1976 dollars by multiplying by an inflation rate of 1.856 taken from Cotton's [4] computer analysis for ships. The resulting cost of the machinery acquisition was $61,621.00 per induced-draft fan. This corresponds

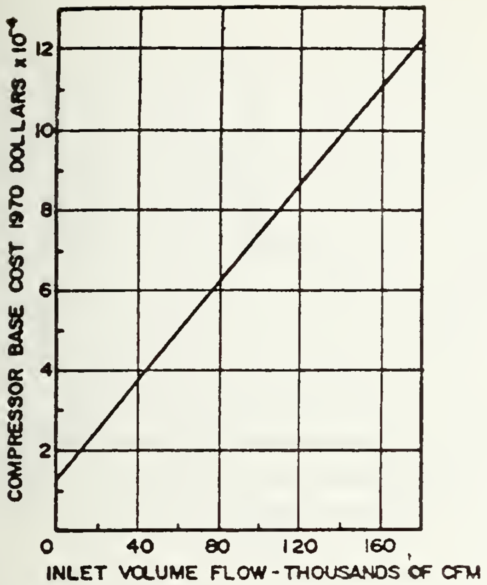


Fig. 18 Base cost of compressors [6]

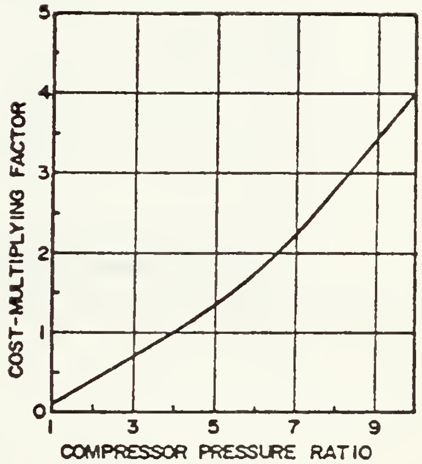


Fig. 19 Effect of pressure ratio on compressor cost [6]

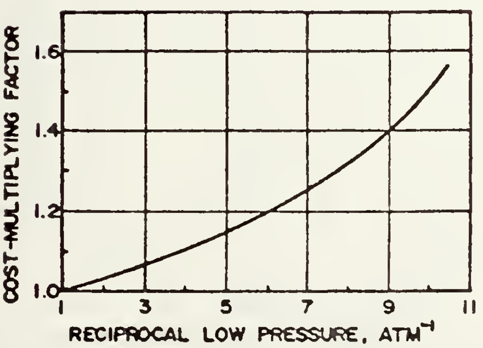


Fig. 20 Effect of vacuum conditions on turbine and compressor cost [6]

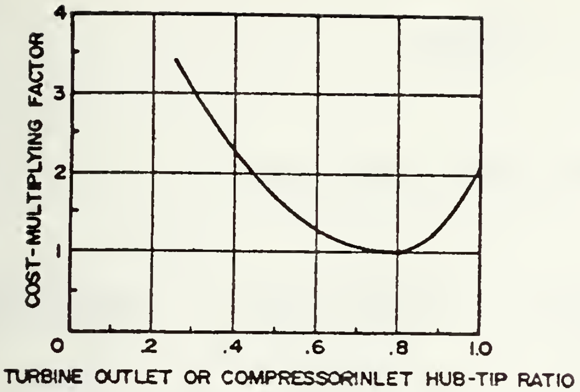


Fig. 21 Effect of low-pressure-end hub-tip ratio on compressor and turbine cost [6]

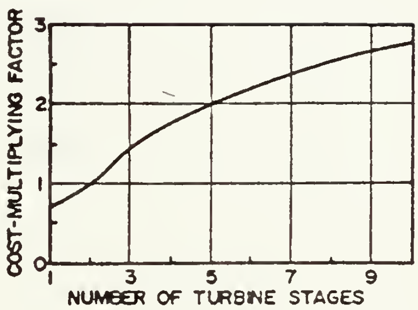


Fig. 22 Effect of number of stages on compressor and turbine cost [6]

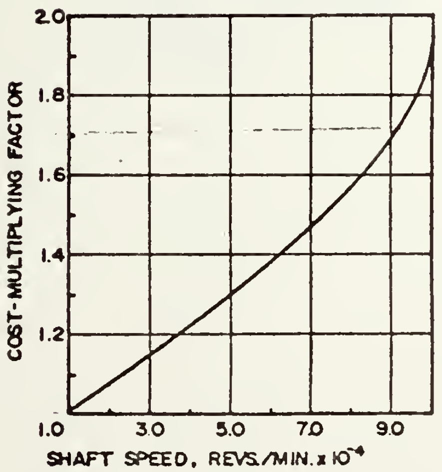


Fig. 23 Effect of shaft speed on compressor and turbine cost [6]

TABLE 7

MACHINERY ACQUISITION COST

PARAMETER	FAN DESIGN	FIGURE	FACTOR
CFM	79,500	19	$62,000
PR	1.88	20	0.35
R_h/R_t	0.75	21	1.00
ATM^{-1}	1.84	22	1.02
# STAGES	3	23	1.50
RPM	9084	24	1.00

COST	$33,201
INFLATION RATE	1.856
MACHINERY ACQUISITION COST/GENERATOR	$61,621
TOTAL MACHINERY ACQUISITION COST PER SHIP	$184,863

to $184,863 per ship, since there are three generators on the ship.

3.3.2. SHIP IMPACT COSTS

The method of marginal cost factors developed by Sejd [19] is suited for use in estimating the ship-system cost resulting from adding components such as the induced-draft fan. Marginal cost is defined as the "cost" of one additional unit of a design parameter at some specified level. The influence of the additional unit is measured by its direct influence on manning, electric power, space and weight.

One of the assumptions necessary to use marginal cost is a "tight" ship design. That is, there is no excess weight, space or power margin in the design. Additionally, the weight of the induced-draft fan and supporting structure was estimated at two tons, based on the similar characteristics between the fan and forced-draft blowers currently in use as discussed in reference 17. The total weight addition to the ship resulting from adding one induced-draft fan to each of the three gas-turbine generators is six tons.

Updated marginal cost factors from Howell [11] were used for this analysis. The marginal cost factor for the addition of one ton located 20 feet below the main

deck is $5,867/ton. The resulting cost to the ship of adding 6 tons of induced-draft fans, therefore, is $35,202. The induced-draft fans will be located inside the exhaust ducting presently existing for the gas turbines. It is therefore estimated that there will be no additional space cost. Additionally, since there should be no additional watch or control operator required for the fans because they will be enclosed in the ducting, there will be no manning cost increase. Finally, the electrical marginal cost factor is zero since the induced-draft fan can be driven directly by the gas turbine. The net cost results from weight only and totals $35,202.

3.3.3. OPERATIONAL RUN COSTS

There were five options available for the power and efficiency utilization in the electrical portion of the cost/performance option tree in Figure 17. These five options will each be considered and the direct run cost calculated for each.

3.3.3.1. CONSTANT PERFORMANCE/DECREASE COST - PROPULSION OR ELECTRICAL POWER

The first option calls for constant ship-system performance with a decrease in ship impact. This can be accomplished by decreasing the fuel consumption for the

Allison gas turbine with the inverted Brayton cycle while maintaining the baseline demand for electrical power. Because the inverted Brayton cycle has increased the net power of the gas turbine without increasing the fuel consumption, a decrease in the demand for this potential results in a direct decrease in gas-turbine power. In other words, for the relatively small quantities of power being considered, the inverted Brayton cycle provides the final portion of the power demanded and the gas turbine does not have to "work" as hard as it would without the induced-draft fan. The fuel savings could be considered as either affecting the propulsion of electrical power generation areas since they are both served from the same storage tanks.

There are three factors which affect the direct run cost in this application. The rate of fuel consumption of the gas turbine, the cost of the fuel, and the time of operation determine the savings in fuel cost.

The fuel consumption rate can be computed from standard data for gas-turbine operation as found in Figures 24 and 25 for the Allison gas turbine. The gas-turbine power to enter the figures was calculated as 6.5% under the baseline power demand of 1135 kw for each generator. The resulting savings using the inverted Brayton cycle is 160 lbm/hr. Table 8 summarizes this step as well as the remaining steps to find the cost savings.

FIGURE 24

GAS GENERATOR OUTLET TEMPERATURE

GENERATOR SPEED	1200 rpm
ENGINE SPEED	13820 rpm
INLET LOSS	4" water
OUTLET LOSS	6" water
GENERATOR EFFICIENCY	95.8%
GEARING EFFICIENCY	97.0%
LHV OF FUEL	18200 Btu/#

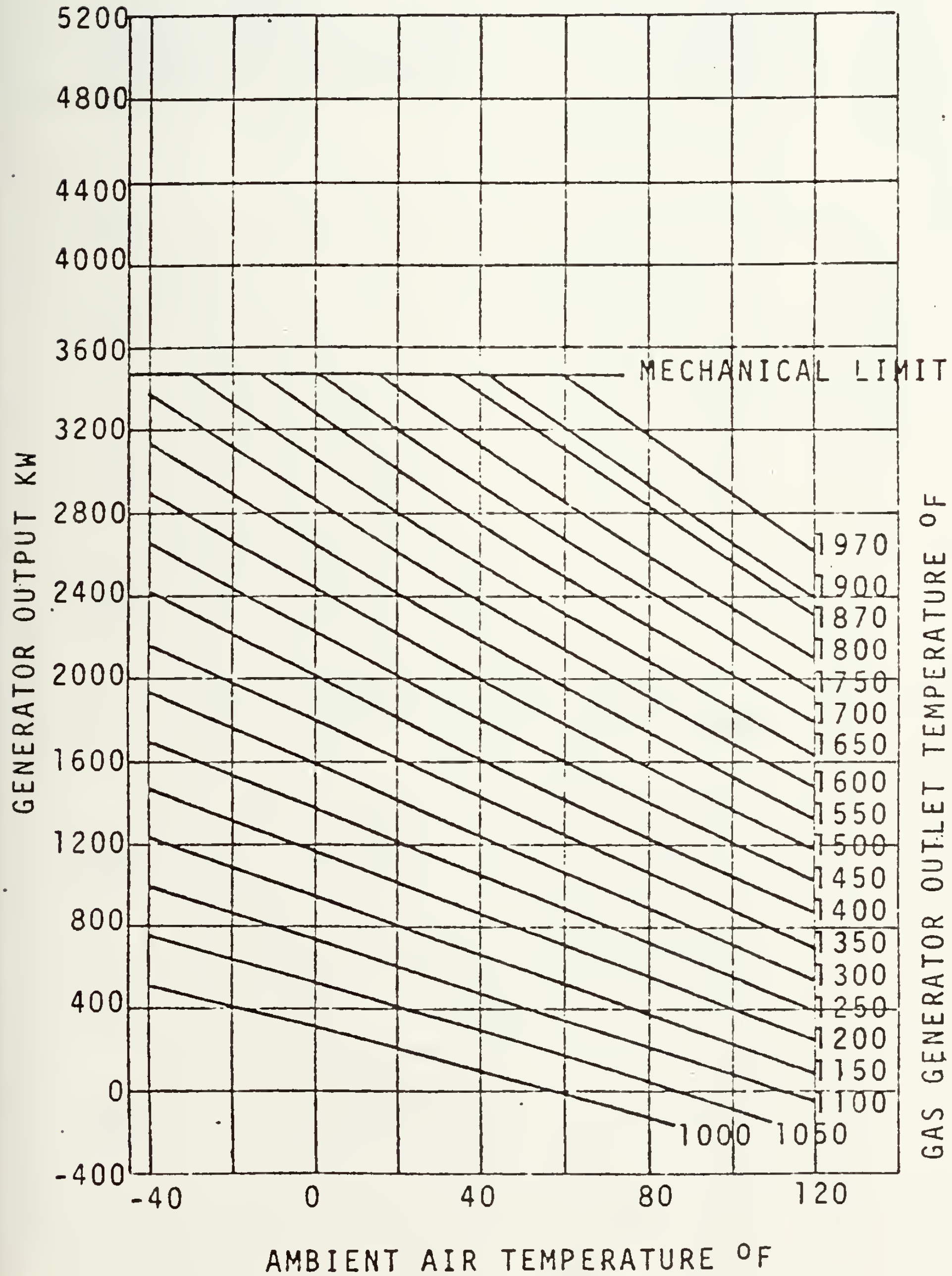
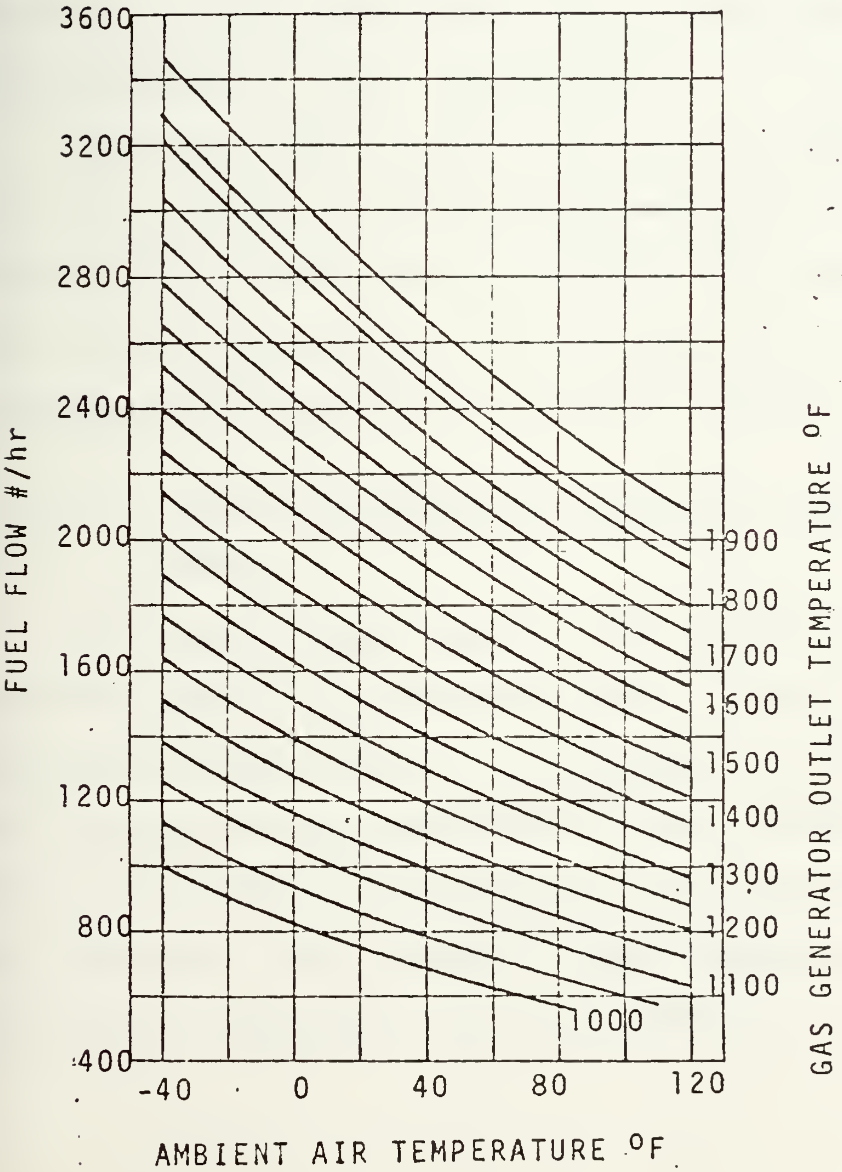


FIGURE 25

FUEL FLOW

GENERATOR SPEED	1200 rpm
ENGINE SPEED	13820 rpm
INLET LOSS	4" water
OUTLET LOSS	6" water
GENERATOR EFFICIENCY	95.8%
GEARING EFFICIENCY	97.0%
LHV OF FUEL	18200 Btu/#



The cost of the fuel was estimated to be $13/bbl based on data from MIT course 13.22, Propulsion Systems, and spot checked in recent oil-company journals. No escalation rate has been applied to this price over the 30 year life-cycle of the ship.

The final factor needed to compute fuel cost savings is the time of operation. This would vary from ship to ship and from year to year but a figure of 45% underway time and 20% inport time without shore power is representative of naval combatants.

Combining these factors results in a direct run cost fuel savings of $35,248 per year and an undiscounted value of $1,057,440 over the 30 year life-cycle of the ship. This represents a 6.3% savings on the electrical-power-generation fuel cost as shown in Table 8.

3.3.3.2. CONSTANT PERFORMANCE/DECREASED COST - ELECTRICAL POWER

The baseline ship requires two generators to achieve baseline demand for electrical power. This is standard naval ship design practice as discussed previously. The addition of the induced-draft fan, however, provides sufficient additional horsepower to the gas turbine that the electrical power demand for this baseline could be met with one engine on the line, assuming the generator

TABLE 8

DIRECT RUN COSTS -

FUEL SAVINGS WITH TWO GENERATORS

CONDITION	BASELINE SHIP	WITH FAN
Ship Demand	2270 kw	2270 kw
Gas-Turbine Load	1135 kw	1061 kw
Ambient Temperature	100°F	100°F
Turbine Inlet Temp (Fig 24)	1440°F	1380°F
Fuel Flow (Fig 25)	1260 lbm/hr	1180 lbm/hr
Ship Electrical Fuel Rate	2520 lbm/hr	2360 lbm/hr

Fuel Savings	160 lbm/hr
Cost of Oil	(\$13/bbl)·(bbl/336 lbm)
Operation Time	(8760 hrs/yr)·(65% on Line)
Annual Fuel Savings	\$35,248
Life-Cycle Fuel Savings	\$1,057,440

could handle the increase.

The fuel savings operating with one engine was calculated using the same method described in the previous section, again utilizing Figures 24 and 25. The resulting fuel savings is 770 lbm/hr as shown in Table 9. If the ship operated in this condition all the time, the annual savings would be $169,622 and the 30 year life-cycle savings would be $50,890,012. This undiscounted savings is nearly equal to the price of a new naval combatant and represents a 30% savings in the electrical-power-generation fuel cost.

3.3.3.3. CONSTANT PERFORMANCE/DECREASED COST - AUXILIARY AREA

The following application takes more of a "total-energy" approach and proposes utilizing steam heating for the ship-heating requirements instead of the electric heat used on baseline ship. Admittedly, this could be accomplished without the induced-draft fan, but it is presented to demonstrate the completeness of the approach to cost-effectiveness.

The winter heating demand for the baseline ship is defined by Abbott [1] as 543.5 kw. Figures 24 and 25 were again utilized to compute the fuel savings of the ship with steam heat and the induced-draft fan, over

TABLE 9

DIRECT RUN COSTS -

FUEL SAVINGS WITH ONE GENERATOR

CONDITION	BASELINE SHIP	WITH FAN
Ship Demand	2270 kw	2270 kw
Gas-Turbine Load	1135 kw	2000 kw
Ambient Temperature	100°F	100°F
Turbine Inlet Temp (Fig 24)	1440°F	1700°F
Fuel Flow (Fig 25)	1260 lbm/hr	1750 lbm/hr
Ship Electrical Fuel Rate	2520 lbm/hr	1750 lbm/hr

Fuel Savings	770 lbm/hr
Cost of Oil	(\$13/bbl)·(bbl/336 lbm)
Operation Time	(8760 hrs/yr)·(65% on line)
Annual Fuel Savings	\$169,634
Life-Cycle Fuel Savings	\$50,890,012

the baseline ship. The steps shown in Table 10 result in fuel savings of 660 lbm/hr, assuming $40^{o}F$ is the average ambient temperature. Since the ship would only need heat for half of any year, the annual fuel cost savings would be $50,331 and the undiscounted 30 year savings would be $1,509,930. This represents a 9% savings in the electrical-power-generation fuel cost.

3.3.3.4. INCREASING PERFORMANCE - PROPULSION

Utilizing the fuel savings from the electrical-power generation to increase the endurance range of the ship is one method of increasing performance. The fuel savings has previously been calculated at 160 lbm/hr for the electrical system when employing the inverted Brayton cycle.

The fuel savings can all go directly to propulsion since the two systems share common fuel-storage tanks. The propulsion system consists of LM2500 gas turbines. Jane's Fighting Ships [12] lists the endurance as 6000 NM at 20 knots. Using the performance curve for the LM2500 shown in Figure 26 and the 11,130 hp per turbine reported by Rains [17] for the endurance cruise mode, 2 years out of dock in sea state 4, the rate of fuel consumption for the propulsion system is calculated at 10684.8 lbm/hr, as seen in Table 11.

TABLE 10

DIRECT RUN COSTS -

FUEL SAVINGS WITH STEAM HEAT

CONDITION	BASELINE SHIP	STEAM HEAT
Electric Heat Load	543.5 kw	0
Electrical Load	2736 kw	2193 kw
Gas-Turbine Demand	1368 kw	771.1 kw
Ambient Temperature	40^{o}F	40^{o}F
Turbine Inlet Temp (Fig 24)	1325^{o}F	1160^{o}F
Fuel Flow (Fig 25)	1350 lbm/hr	1026 lbm/hr
Ship Electrical Fuel Rate	2700 lbm/hr	2040 lbm/hr

Fuel Savings	660 lbm/hr
Cost of Oil	(\$13/bbl)·(bbl/336 lbm)
Operation Time	(8760 hr/yr)·(22.5% on Line)
Annual Fuel Savings	\$50,331
Life-Cycle Fuel Savings	\$1,509,930

By saving 160 lbm/hr from the electrical system for the 300 hours of endurance time, 48,000 lbm of fuel would be saved. The ship endurance range could then be increased only about 65 miles, or a 1% increase, as seen in Table 11.

3.3.3.5. INCREASING PERFORMANCE - ELECTRICAL POWER

The electrical-power-generation capacity of the baseline system can be increased by the addition of the induced-draft fan to the gas turbine, if the generator is able to handle the increase. This can be accomplished with the induced-draft fan design which resulted in a 9% increase in the gas-turbine rating.

The three ship's-service generators total 6000 kw capacity. An increase in each one of 9% results in a total of 6540 kw. Since the fuel consumption is the same as the baseline, the cost per kw also shows a 9% improvement, 3.106¢ per kw compared to 3.385¢ per kw.

3.3.3.6. INDIRECT RUN COSTS - WEIGHT

The weight change of a subsystem will affect the endurance range of the ship. Basically, an increase in weight will sink the ship deeper, which will increase the frictional resistance and therefore increase the power required to drive the ship at the same speed. Greater power requires greater fuel consumption resulting in increased cost.

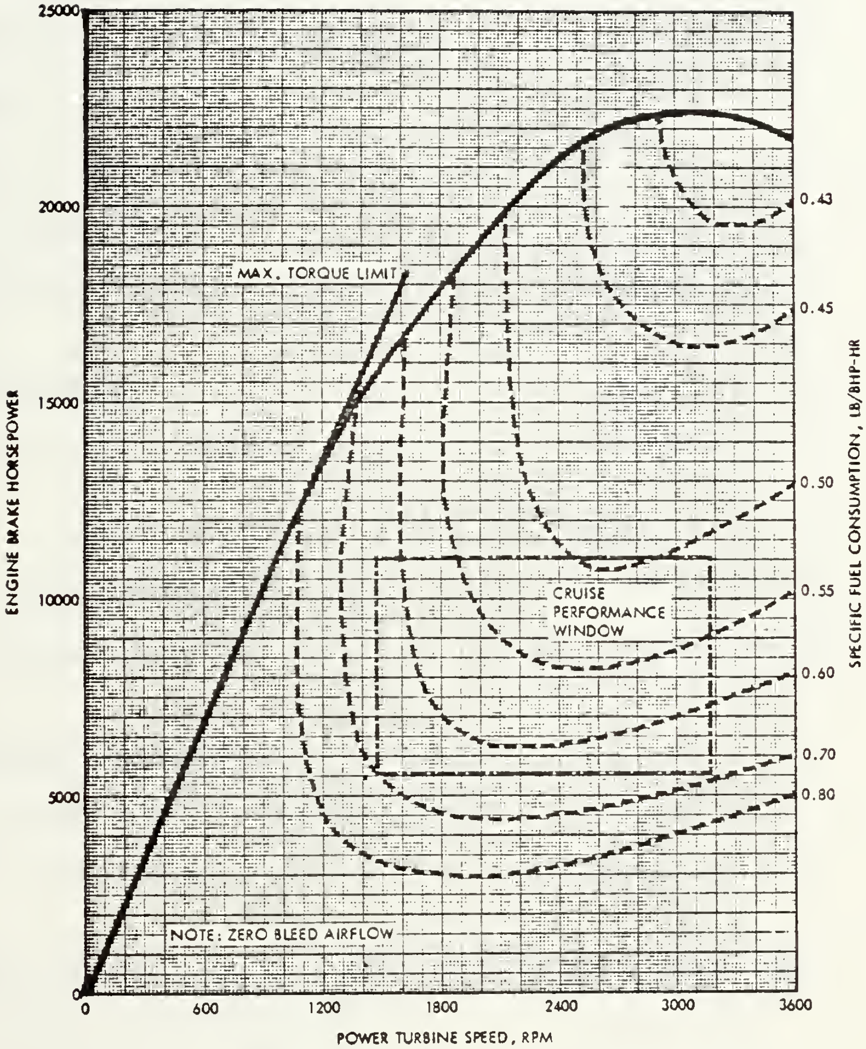


Fig 26 LM2500 PERFORMANCE CURVE [17]

TABLE 11

INCREASE PERFORMANCE -
ENDURANCE

FUEL SAVED	
Fuel Savings (see Table 9)	160 lbm/hr
Endurance Range	6000 NM
Endurance Speed	20 kts
Time (Range/Speed)	300 hrs
Fuel Saved (Time x Savings)	48,000 lbm
FUEL REQUIRED	
HP/Engine Propulsion	11,130 hp
SFC	0.48
Fuel Required/Engine	5,342.4 lbm/hr
Fuel Required/Ship	10,684.8 lbm/hr
INCREASED ENDURANCE	
Fuel Saved/Fuel Required = Time	4.5 hours
Time x Speed = Range	90 miles
ITERATE TO ACCOUNT FOR ELECTRICAL FUEL REQUIRED	
Electrical Fuel Required (Rate x Time)	11340 lbm
Fuel Saved - Elec. Fuel Required	36660 lbm
Fuel Saved/Fuel Required = Time	3.4 hrs
Time x Speed = Range	68 miles

For the electrical-power-generation system, only an induced-draft fan was added. The total weight change to the ship was previously estimated as 6 tons. Table 12 follows the effect of the weight change through to greater cost using the theory just described, assuming the total change can be calculated as a flat-plate, frictional-resistance increase. The corresponding decrease in endurance range is 20 miles, the increase in the annual cost for propulsion fuel is $7,190 per year, and the increase in the total life-cycle fuel cost is $215,721. This represents an increase of ½% in the propulsive fuel cost.

3.3.3.7. INDIRECT RUN COSTS - VOLUME

Fuel storage-tank volume requirements could be decreased in the ship design by decreasing the fuel consumption rate, as is possible with the inverted Brayton cycle on the electrical-power-generation system. The fuel savings have already been computed for endurance range as 48,000 lbm. The density of the fuel is 35 lbm/cu-ft resulting in a volume savings of 1371 cu-ft. The impact on a ship design has been estimated as 3 tons per 1000 cu-ft of tankage volume. This relatively small impact results from the low priority of fuel tanks' locations, and shapes. The resulting volume savings on the ship if the fuel were removed, expressed in weight on the ship

TABLE 12

INDIRECT RUN COST - WEIGHT

1.	Tons per Inch Immersion	55 tons/in.
2.	Increase Draft = Weight/TPI	0.11 in.
3.	Increase in Wetted Surface (L x T) $L = (2 \times L_o) = (2 \times 525) = 1050$ ft.	9.6 sq. ft.
4.	Coefficient of Drag (Flat Plate)[18]	2.0
5.	Resistance = $C_D \frac{\rho}{2} S V^2$	21,909 lbf
6.	EHP = Resistance/325.5	67.3 hp
7.	Additional EHP/Engine	33.65 hp
8.	Total EHP/Engine	11,164 hp
9.	SFC (Fig 26)	0.48
10.	Fuel (SFC x HP) x 2 engines	10,717.4 lbm/hr
11.	Fuel Originally (Table 11)	10,684.8 lbm/hr
12.	Increase Fuel Required	32.64 lbm/hr
13.	Fuel Increase (Rate x Time)	9792 lbm
14.	Decrease in Range (Fuel Increase/Fuel Required) (Table 12)	1 mile
15.	Annual Cost (See Table 9)	$7,190
16.	Life-Cycle Cost	$215,721

design, is 1.37 tons. The procedure followed for the weight change would reflect an increase in endurance range of 4 miles, an annual savings in propulsive fuel of $1557 and a total life-cycle fuel cost savings of $46,739.

3.3.4. OPERATIONAL MANNING COSTS

The cost of operating the ship will generally increase as machinery is added to the ship, as described in section 3.2.1.1. For the case of the induced-draft fan, however, it is estimated that no additional men are required because the machinery is located inside the exhaust ducting and is controlled directly by the gas turbine. There might be a requirement for checking bearing temperatures periodically, or even changing oil, but it is not felt that the addition of any personnel is required for these items.

3.3.5. OPERATIONAL MAINTENANCE COSTS

The operational maintenance cost should be calculated for machinery based on statistical information such as failure rates and time to repair data. Since this information is not available for the induced-draft fan, no attempt is made to estimate the operational maintenance costs.

3.3.6. SUMMARY OF COSTS

The sum of the machinery acquisition, ship impact, and operational manning, maintenance, and run costs is presented in Table 13 for the varying performance and impact options. This cost represents the "true cost" of the system, defined previously as the total impact of the system on the total ship.

The alternative with the greatest cost savings is the proposal which operates on one ship's-service generator. This operational mode was made possible because the induced-draft fan added just enough power to meet the baseline demand. The cost savings for this alternative was an order of magnitude greater than any other option and represented a 30% savings in the fuel consumption for generating electrical power. However, it is not standard operating practice to run just one generator except in the merchant fleet. It is not up to the ship designer to change standard operating procedures on his own judgement, so this alternative must be considered as nice to know, but not applicable.

The option of shifting the design from electric to steam heat was possible before adding the induced-draft fan. This total-energy approach option resulted in savings just over one-million dollars over the life cycle of the ship.

TABLE 13

SUMMARY OF LIFE-CYCLE COSTS

COST CHANGE	DECREASE FUEL CONSUMPTION	ONE GENERATOR	STEAM HEAT	INCREASE RANGE	INCREASE KW
MACHINERY ACQUISITION	+$184,863	+$184,863	+$184,863	+$184,683	+$184,683
MARGINAL WEIGHT	+$35,202	+$35,202	+$35,202	+$35,202	+$35,202
MARGINAL VOLUME	0	0	0	0	0
MARGINAL MANNING	0	0	0	0	0
MARGINAL ELECTRICAL	0	0	0	0	0
DIRECT RUN	-$1,057,440	-$50,890,012	-$1,509,930	-	-
INDIRECT RUN	+$215,721	+$ 215,721	+$215,721	+$215,721	+$215,721
INDIRECT VOLUME	+$46,739	+$46,739	+$46,739	+$46,739	+$46,739
MANNING	0	0	0	0	0
NET COST	-$574,915	-$50,407,487	-$1,027,405	+$482,525	+$482,525
PERFORMANCE CHANGE				+65 NM	+540 KW

The addition of the induced-draft fans to the electrical-power-generation system results in a total cost savings of over $600,000 for the life-cycle cost. The 9% savings in the cost of generating the electrical power was directly the result of utilizing the inverted Brayton cycle.

The option for increasing the endurance range of the ship cost nearly one-half million dollars over the life-cycle of the ship but only gains 1% in endurance range. The results for this option are so low because of the high demand of fuel of the propulsion gas turbines relative to the savings of the induced-draft fan on the much smaller electrical-power-generation gas turbine.

The final option increased the ship's electrical capacity by 9% for an acquisition cost increase of $150,000 and a life-cycle cost increase just over $400,000.

The various options are not quite ready to be evaluated at this point. The other factor in measuring naval ship systems, performance, has not been measured. But the costs of the various options are seen from the results of Table 13 and positive gains in quantity as well as percentage are evident.

The numerical accuracy of the results can be justly contested. Costs are notoriously difficult to estimate

and the areas where these calculations have been made are subject to far-ranging variations. Ships at sea are subject to wind, waves, currents and varying load conditions. Machinery costs cannot be precisely measured at this stage of the design. Many assumptions had to be made to complete the analysis and they are listed in Table 14 to allow the reader to obtain a feel for the degree of accuracy in the results. The cost estimates are, therefore, crude and open to question. An attempt has been made to be conservative in most respects and it is felt that these results do provide a viable starting point for a more advanced study.

3.3.7. PERFORMANCE FACTORS

The factors for systems-engineering performance are qualatative with respect to the cost-effectiveness of the options. Though the factors do have quantatative values such as noise level, availability percentage, and degree of damage, these levels are, in effect, constraints on the ship designer. It is not possible to place a dollar value on a systems-engineering performance level which is less than the established limit. Therefore, the systems-engineering performance factors are posed as options for the ship designer. It is engineering judgement that will decide if a fuel savings resulting from additional machinery is worth the increase in noise

TABLE 14

LIST OF ASSUMPTIONS FOR COSTING

1. Weight of Induced-Draft Fan is approximately equal to Forced-Draft Blower.
2. No ship volume is required for Induced-Draft Fan other than exhaust ducting.
3. No increase in manning is required for Induced-Draft Fans.
4. Induced-Draft Fan run by direct drive from gas turbine.
5. Cost of fuel is constant for next 30 years.
6. Annual underway percentage is 45%.
7. Annual inport percentage without electrical shore power is 20%.
8. Fuel savings are constant for other operating conditions.
9. Generator on board is large enough to handle 9% increase.
10. Average ambient temperature in winter is 40°F.
11. Flat-Plate resistance approximates increase in draft.
12. 1000 cu. ft. of tankage volume converts to 3 tons weight.

level, for instance, as long as it is below the limit set for noise.

The systems-engineering performance factors have been listed in Table 15 with an engineering judgement and justification as to their impact on the ship for adding the induced-draft fans to complete the inverted Brayton cycle. None of the factors are felt to exceed their design constraints and are, therefore, satisfactory.

3.4. CONCLUSIONS ON CYCLE UTILIZATION

The conclusions on the approach to measuring the cost-effectiveness of the cycle utilization will be divided into the method of analysis and the results concerning the inverted Brayton cycle.

3.4.1. SUMMARY OF METHODOLOGY

A technique has been presented to evaluate any subsystem alternative on naval ships. This technique investigates the ship as a system and identifies all of the performance and cost factors necessary to evaluate naval ship systems.

The cost factors include acquisition and life-cycle costs. The acquisition costs are composed of the system machinery costs and the weight, space, manning and electrical costs to the ship resulting from the machinery. The life-cycle costs, called operational costs, include

TABLE 15

SYSTEMS-ENGINEERING FACTORS
RELATIVE IMPACT ON OPTIONS

FACTOR	DECREASE FUEL CONSUMPTION	ONE ELECTRICAL GENERATOR	STEAM HEAT	INCREASE RANGE	INCREASE KW
AVAILABILITY	NO CHANGE	NO CHANGE	DOWN (MAINTENANCE)	NO CHANGE	NO CHANGE
STANDARDIZATION	DOWN SLIGHTLY (FAN)	DOWN SLIGHTLY (FAN)	NO CHANGE	DOWN SLIGHTLY (FAN)	NO CHANGE
RISK	UP SLIGHTLY (FAN)	UP SLIGHTLY (FAN)	NO CHANGE	UP SLIGHTLY (FAN)	UP SLIGHTLY (FAN)
SURVIVABILITY	NO CHANGE	DOWN (LOSS OF POWER)	DOWN (NO BYPASS)	NO CHANGE	NO CHANGE
SIGNATURE	UP SLIGHTLY (EXH. TEMP.)	UP SLIGHTLY (EXH. TEMP.)	NO CHANGE	UP SLIGHTLY (EXH. TEMP.)	UP SLIGHTLY (EXH. TEMP.)
FLEXIBILITY	NO CHANGE	NO CHANGE	DOWN (NO ALTERNATIVE)	NO CHANGE	UP SLIGHTLY (1 GEN. OPS)

the direct and indirect costs of operating, manning, and maintaining the machinery.

An option tree for identifying possible uses of gains in power or efficiency has been described. The source of energy for the options was propulsion or electrical-power generation. The integration of these options represents the total-ship system approach to the design.

The value of the ship-system approach is that all of the factors involved in analyzing alternative systems are presented in a logical, methodical manner. It is this type of approach which allows the ship designer to make intelligent decisions.

3.4.2. CONCLUSIONS ON THE INVERTED BRAYTON CYCLE STUDIES

The electrical-power-generation application of the inverted Brayton cycle showed promising gains in power and efficiency because the steam in the baseline waste-heat boiler was not converted to electrical power, but was used elsewhere as heat energy. Five options were investigated for utilizing the power and efficiency gains.

The first option utilized the efficiency of the cycle to decrease the consumption rate of the fuel. This application resulted in a life-cycle cost savings of almost $600,000. The performance factors remained

nearly constant. Therefore, the cost-effectiveness of this alternative is positive. The ship designer should be interested in the 9% electrical generation fuel consumption savings from the viewpoint of resource management, but the degree of life-cycle cost savings is small compared to the acquisition cost of the ship ($50,000,000).

One of the alternatives investigated required a change in standard operating practices. The current philosophy is that naval ships should operate with at least two generators on the line as a precautionary procedure. But if just one generator were allowed on the line, the addition of the induced-draft fan would permit sufficient power to be generated to meet the electrical power demand for the case of the baseline ship. A considerable fuel savings results from reducing the number of generators on the line, even while maintaining performance at a constant. The results of the life-cycle cost studies showed a savings over $50,000,000. This is certainly a significant number to the ship designer, as well as the 30% savings in electrical-power-generating fuel consumption.

The last option studied which maintained performance nearly constant was shifting from electric heat to steam heat for the ship. This was possible before the induced-

draft fan was added to the system because the waste-heat boiler was part of the baseline design. This "total-energy" option was investigated anyway to demonstrate a possible integration of the option tree. The gains from this option over the life-cycle of the ship are just over $1,000,000. Operational performance was not altered, but some of the systems-engineering performance factors were degraded. Engineering judgement on this alternative estimates the cost-effectiveness of the option is slightly positive.

An attempt to improve the operational performance of the baseline ship in the propulsive area by increasing the endurance range with fuel saved from the electrical system resulted in an insignificant gain of 1% in endurance. This option definitely shows negative cost-effectiveness.

The final option studied also worked on improving the operational performance of the baseline ship. In this case the electrical-power capacity of the ship was increased with the addition of the induced-draft fans. This increase added 540 kw to the ship's capacity and represented a 9% gain. This is the type of result the ship designer likes to have in his "back pocket" when nearing the completion of preliminary design.

It is proper at this time to point out that the addition of the induced-draft fans to the electrical-power-generation system allows the ship designer the option of saving 9% of the electrical-power fuel consumption from the start, and later, when the electrical demand grows, the growth capacity is already installed. Even so, one would have to say that only one of the options has a clear-cut positive cost-effectiveness. That option was operating with one generator on the line, and is not in keeping with standard operating practice. The cost of fuel makes the inverted Brayton cycle attractive, but that cost is not yet high enough to insure positive cost-effectiveness on a naval ship.

The application of the inverted Brayton cycle to COGAS propulsion systems should not be considered by the naval ship designer. The power gains from the gas turbines in the electrical application result because the thermodynamic cycle is different from the COGAS cycle. It was shown that the steam turbine will lose power as a result of adding an induced-draft fan to the propulsion cycle. The combined steam-turbine and gas-turbine power will not improve when adding an inverted Brayton cycle to a COGAS system.

CHAPTER 4 FINDINGS, CONCLUSIONS, AND RECOMMENDATIONS

4.1. SUMMARY OF FINDINGS

The current "energy crisis" highlights the need for innovation in fuel economy measures for new naval ship designs. These innovations must be consistent with the overall ship design philosophy for the ship and should be measured with the total-ship impact in mind.

One such innovation, the inverted Brayton cycle, showed good promise for industrial and merchant ship applications. Since gas turbines will play a prominent role in the future of the Navy's non-nuclear fleet, an investigation into the cost-effectiveness of the cycle applied to propulsion and electrical power generation gas turbines is worthwhile.

The baseline electrical-power-generation system investigated was the Allison gas turbine and DD-963 waste-heat boiler. The waste-heat steam is used for auxiliary heat and is not incorporated in the cycle. The addition of the induced-draft fan increased gas turbine power and efficiency about 9%.

Several options were investigated to utilize the power and efficiency gains from the electrical-power-generation application of the inverted Brayton cycle. The only option which showed a clear-cut positive cost-

effectiveness was not in keeping with standard operating practice. In this option, the addition of the inverted Brayton cycle provided sufficient additional power to meet the electrical-power demand on just one engine, with no change in the operational performance factors (assuming the generator was sized properly). The life-cycle cost savings using this option totaled over $50,000,000 which is on the same order of magnitude as the acquisition of a small combatant.

Other options, which could be combined to enhance their payoff, included decreasing fuel consumption while maintaining performance factors at a constant, or increasing the operational performance of the electrical power capacity with an accompanying increase in cost. These options both showed slight increases in life-cycle cost-effectiveness, but neither could be said to be dramatic gains. Since they were achieved in the same manner, the addition of an induced draft fan, either option is available to the ship if the machinery is added. This combination of options increases their desirability.

The only option which clearly had a negative cost-effectiveness was the utilization of the electrical-power generation fuel savings to increase the endurance range of the ship. In this case, the fuel consumption

rate of the propulsive plant is so much greater than the savings from the much smaller electrical-power-generation gas turbine that the gain was insignificant.

A final option, employing more of a total-energy approach, used waste-heat steam to meet the space-heating demand instead of the electrical heat used by the baseline ship. Because this option was available without the induced-draft fan, it is not a real application of the inverted Brayton cycle. But the analysis to demonstrate the completeness of the methodology resulted in life-cycle cost savings over $1,000,000. Because systems-engineering performance factors decreased with this option, it could be said that the cost-effectiveness was only slightly positive.

4.2. ASSUMPTIONS AFFECTING VALIDITY

Many assumptions were necessary to reach the results stated above. A list of the cost assumptions was presented in Table 15 along with assumptions concerning thermodynamics, found in Tables 2, 3, and 4. In addition to the assumptions, the method of costing machinery and determining fuel consumption is subject to variables beyond the scope of this study as discussed in section 3.3.6. However, by making conservative estimations and by being complete in the scope of the analysis, the results are felt to be a true indication of the relative merit of the options.

4.3. CONCLUSIONS

The proposal of adding an inverted Brayton cycle to the electrical power generation system is marginally cost-effective. There are increases in power or efficiency and they can be turned into worthwhile fuel savings, but the present cost of fuel is not high enough to clearly offset the 'true cost' to the ship system, unless standard operating practices are allowed to change.

If fuel conservation and resource management become the dominant forces in ship design philosophy, the ship designer could consider the use of the inverted Brayton cycle, perhaps in concert with other economy measures, to be significantly cost-effective.

4.4. RECOMMENDATIONS FOR FUTURE WORK

The original assumption in this study was to limit the thermodynamic analysis to one climatic and load condition in order to investigate the potential of the inverted Brayton cycle in naval ship applications. The electrical-power-generation gas turbine has shown sufficient power gains that a more detailed analysis of the gas turbine off-design points should be completed to give a more comprehensive analysis. This should be integrated with varying climatic conditions for a complete thermodynamic picture.

There is no potential for the propulsive application of the inverted Brayton cycle and, therefore, no additional research is necessary for the LM2500 propulsion gas turbine.

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APPENDIX A

CALCULATIONS FOR GAS-TURBINE CYCLE

PRESSURE CALCULATIONS

BRAYTON CYCLE

P_o	Atmospheric
P_1	P_o - Intake Losses
P_2	$P_1 \times PR$
P_3	$P_2 - \Delta P_{2\rightarrow 3}$
P_4	P_o + Exhaust Losses

BRAYTON CYCLE WITH WASTE-HEAT BOILER

$P_o - P_3$	Same as Brayton Cycle
P_{5a}	P_o + Exhaust Losses
P_{4a}	$P_{5a} + \Delta P_{4\rightarrow 5}$

INVERTED BRAYTON CYCLE

$P_o - P_3$	Same as Brayton Cycle
P_{6b}	P_o + Exhaust Losses
P_{5b}	$P_{4b} - \Delta P_{4\rightarrow 5}$
P_{4b}	Specified

Subscripts refer to Figures 1,2, &3.

ENTHALPY CALCULATIONS

BRAYTON CYCLE

h_1:	T_o	Specified
	T_1	T_o
	P_{r1}	K&K [13] Table 1 using T_1
	h_1	K&K Table 1 using T_1
h_3:	T_3	Specified
	P_{r3}	K&K Table 4 using T_3
	h_3	K&K Table 4 using T_3
h_2:	P_{r2}	$P_{r1} \times P_2 / P_1$
	h_{2is}	K&K Table 1 using P_{r2} (MW = 28.9)
	Δh_{2is}	$h_{2is} - h_1$
	Δh_{2-1}	$\Delta h_{2is} / \eta_{is_c}$
	η_{is_c}	Specified
	h_2	$h_1 + h_{2\rightarrow1}$
h_4:	P_{r4}	$P_{r3} \times P_4 / P_3$
	h_{4is}	K&K Table 4 using P_{r4} (MW = 28.9)
	Δh_{4is}	$h_3 - h_{4is}$
	Δh_{3-4}	$\Delta h_{4is} \times \eta_{is_t}$
	η_{is_t}	Specified
	h_4	$h_3 - \Delta h_{3\rightarrow4}$

ENTHALPY CALCULATIONS

BRAYTON CYCLE WITH WASTE-HEAT BOILER

h_1, h_2, & h_3		Same As Brayton Cycle
h_{4a}:	P_{r4a}	$P_{r3} \times P_{4a} / P_3$
	$h_{4a_{is}}$	K&K Table 4 using P_{r4a} (MW = 28.9)
	Δh_{is}	$h_3 - h_{4a_{is}}$
	Δh_{3-4a}	$\Delta h_{is} \times \eta_{is_t}$
	h_{4a}	$h_3 - \Delta h_{3-4a}$
h_{5a}:	ΔT_{whb}	Specified
	T_{5a}	$T_{4a} - \Delta T_{whb}$
	h_{5a}	K&K Table 4 using T_{5a} (MW = 28.9)

ENTHALPY CALCULATIONS

INVERTED BRAYTON CYCLE

h_1, h_2, & h_3		Same As Brayton Cycle
h_{4b}:	P_{4b}	Specified
	P_{r4b}	$P_{r3} \times P_{4b} / P_3$
	$h_{4b_{is}}$	K&K Table 4 using P_{r4b} (MW = 28.9)
	Δh_{is}	$h_3 - h_{4b_{is}}$
	Δh_{3-4b}	$\Delta h_{is} \times \eta_{is_t}$
	h_{4b}	$h_3 - \Delta h_{3-4b}$
h_{5b}:	ΔT_{whb}	Specified
	T_{5b}	$T_{4b} - \Delta T_{whb}$
	h_{5b}	K&K Table 4 using T_{5b} (MW = 28.9)
h_{6b}:	P_{r6b}	$P_{r5b} \times P_{6b} / P_{5b}$
	$h_{6b_{is}}$	K&K Table 4 using P_{r6b} (MW = 28.9)
	Δh_{is}	$h_{6b_{is}} - h_{5b}$
	Δh_{5b-6b}	$\Delta h_{is} / \eta_{is_c}$
	h_{6b}	$h_{5b} + \Delta h_{5b-6b}$

FLUID FLOW CALCULATIONS

$\dot{m}_{total}$	Specified
LHV	Specified
$\dot{m}_f/\dot{m}_a$	$(h_3 - h_2)/(LHV - h_3)$
$\dot{m}_a$	$\dot{m}_{total} / (1 + \dot{m}_f/\dot{m}_a)$
$\dot{m}_f$	$\dot{m}_{total} - \dot{m}_a$

POWER CALCULATIONS

$\dot{W}_c$	$\dot{m}_a (h_2 - h_1)$

BRAYTON CYCLE

$\dot{W}_t$	$\dot{m}_{total} (h_3 - h_4)$
hp	$\dot{W}_t - \dot{W}_c$

BRAYTON CYCLE
WITH WASTE-HEAT BOILER

$\dot{W}_{t_a}$	$\dot{m}_{total} (h_3 - h_{4a})$
hp	$\dot{W}_{t_a} - \dot{W}_c$

INVERTED BRAYTON CYCLE

$\dot{W}_{t_b}$	$\dot{m}_{total} (h_3 - h_{4b})$
hp	$\dot{W}_{t_b} - \dot{W}_c - \dot{W}_{idf}$
$\dot{W}_{idf}$	$\dot{m}_{total} (h_{6b} - h_{5b})$

THERMAL EFFICIENCY

η_{th}	hp / $\dot{Q}_h$
$\dot{Q}_h$	$\dot{m}_{total} (h_3 - h_2)$

SPECIFIC FUEL CONSUMPTION

SFC	$\dot{m}_f$ / hp

GAS-TURBINE POWER

TABLE OF RESULTS

LM2500

P_1	14.56	PSIA
P_2	244.61	PSIA
P_3	229.93	PSIA
P_4	15.06	PSIA
T_0	100	°F
T_1	100	°F
h_1	133.55	$\frac{BTU}{LBm}$
P_{R1}	1.5712	
T_3	2150	°F
h_3	693.43	$\frac{BTU}{LBm}$
P_{R2}	26.3464	
h_{2is}	299.22	$\frac{BTU}{LBm}$
h_2	328.46	$\frac{BTU}{LBm}$
P_{R4}	38.860	
h_{4is}	334.26	$\frac{BTU}{LBm}$
h_4	384.54	$\frac{BTU}{LBm}$
$\dot{m}_a$	125.985	$\frac{LBm}{SEC}$
$\dot{m}_f$	2.626	$\frac{LBm}{SEC}$
$\dot{W}_c$	34735	hp

GAS-TURBINE POWER

TABLE OF RESULTS

LM 2500

P_{4b}	6	6	6	6	PSIA
P_{5b}	5.76	5.76	5.76	5.76	PSIA
PR_{4-5}	2.615	2.615	2.615	2.615	
P_{r4}	15.48	15.48	15.48	15.48	
h_{4bis}	258.57	258.57	258.57	258.57	BTU/LBm
h_{4b}	319.45	319.45	319.45	319.45	BTU/LBm
ΔT	500	600	650	700	
T_{5b}	328.3	228.3	178.3	128.3	°F
h_{5b}	191.39	166.66	154.36	142.10	BTU/LBm
P_{r5b}	5.310	3.258	2.488	1.859	
P_{r6b}	13.883	8.518	6.506	4.861	
h_{6bis}	250.94	218.46	202.71	186.67	BTU/LBm
h_{6b}	261.45	227.60	211.24	194.54	BTU/LBm
$\dot{W}_{IDF}$	12745	11086	10347	9539	hp
$\dot{W}_{T}$	68037	68037	68037	68037	hp
$\dot{W}_{NET}$	20556	22215	22954	23762	hp
$\dot{Q}_{H}$	46939.2	46939.2	46939.2	46939.2	BTU/SEC
η_{TH}	0.3096	0.3346	0.3457	0.3578	

GAS-TURBINE POWER

TABLE OF RESULTS

LM 2500

P_{4b}	8	8	8	8	PSIA
P_{5b}	7.68	7.68	7.68	7.68	PSIA
PR_{4-5}	1.961	1.961	1.961	1.961	
P_{r4}	20.64	20.64	20.64	20.64	
h_{4bis}	280.38	280.38	280.38	280.38	BTU/LBm
h_{4b}	338.21	338.21	338.21	338.21	BTU/LBm
ΔT	500	600	650	700	
T_{5b}	398.9	298.9	248.9	198.9	°F
h_{5b}	209.26	184.09	171.74	159.42	BTU/LBm
P_{r5b}	7.282	4.627	3.622	2.786	
P_{r6b}	14.280	9.073	7.104	5.463	
h_{6bis}	252.92	222.66	207.82	192.94	BTU/LBm
h_{6b}	260.63	229.47	214.19	198.85	BTU/LBm
$\dot{W}_{IDF}$	9345	8256	7722	7174	hp
$\dot{W}_{T}$	64624	64624	64624	64624	hp
$\dot{W}_{NET}$	20543	21632	22166	22715	hp
$\dot{Q}_{H}$	46939.2	46939.2	46939.2	46939.2	BTU/SEC
η_{TH}	0.3094	0.3258	0.3338	0.3421	

GAS-TURBINE POWER

TABLE OF RESULTS

LM 2500

P_{4b}	10	10	10	10	PSIA
P_{5b}	9.60	9.60	9.60	9.60	PSIA
PR_{4-5}	1.569	1.569	1.569	1.569	
P_{r4}	25.80	25.80	25.80	25.80	
$h_{4b_{is}}$	298.38	298.38	298.38	298.38	$\frac{BTU}{LBm}$
h_{4b}	353.67	353.67	353.67	353.67	$\frac{BTU}{LBm}$
ΔT	500	600	650	700	
T_{5b}	456.5	356.5	306.5	256.5	°F
h_{5b}	223.45	198.41	185.98	173.61	$\frac{BTU}{LBm}$
P_{r5b}	9.193	6.031	4.798	3.764	
P_{r6b}	14.422	9.461	7.529	5.905	
$h_{6b_{is}}$	253.64	225.28	211.23	197.23	$\frac{BTU}{LBm}$
h_{6b}	258.96	230.03	215.69	201.39	$\frac{BTU}{LBm}$
$\dot{W}_{IDF}$	6462	5752	5405	5055	hp
$\dot{W}_{T}$	61811	61811	61811	61811	hp
$\dot{W}_{NET}$	20614	21324	21671	22021	hp
$\dot{Q}_{H}$	46939	46939	46939	46939	$\frac{BTU}{SEC}$
η_{TH}	0.3105	0.3212	0.3264	0.3317	

GAS-TURBINE POWER

TABLE OF RESULTS

LM 2500

P_{4b}	12	12	12	12	PSIA
P_{5b}	11.52	11.52	11.52	11.52	PSIA
PR_{4-5}	1.307	1.307	1.307	1.307	
P_{r4}	30.96	30.96	30.96	30.96	
h_{4bis}	313.86	1313.86	313.86	313.86	BTU/LBm
h_{4b}	367.00	367.00	367.00	367.00	BTU/LBm
ΔT	500	600	650	700	
T_{5b}	505.9	405.9	355.9	305.9	°F
h_{5b}	235.97	210.75	198.26	185.83	BTU/LBm
P_{r5b}	11.158	7.468	6.015	4.785	
P_{r6b}	14.587	9.763	7.862	6.255	
h_{6bis}	254.44	227.28	213.83	200.46	BTU/LBm
h_{6b}	257.70	230.19	216.58	203.04	BTU/LBm
$\dot{W}_{IDF}$	3953	3538	3333	3132	hp
$\dot{W}_{T}$	59386	59386	59386	59386	hp
$\dot{W}_{NET}$	20698	21113	21318	21519	hp
$\dot{Q}_{H}$	46939	46939	46939	46939	BTU/SEC
η_{TH}	0.3117	0.3180	0.3211	0.3241	

GAS-TURBINE POWER

TABLE OF RESULTS

LM2500

P_{4b}	15.66	15.69	15.69	15.69	PSIA
P_{5b}	15.06	15.06	15.06	15.06	PSIA
PR_{4-5}	1.0	1.0	1.0	1.0	
P_{r4}	40.486	40.486	40.486	40.486	
h_{4bis}	338.06	338.06	338.06	338.06	BTU/LBm
h_{4b}	387.81	387.81	387.81	387.81	BTU/LBm
ΔT	500	600	650	700	
T_{5b}	582.5	482.5	432.5	382.5	°F
h_{5b}	255.4	230.0	217.4	204.9	BTU/LBm
P_{r5b}					
P_{r6b}					
h_{6bis}					BTU/LBm
h_{6b}					BTU/LBm
$\dot{W}_{IDF}$	0	0	0	0	hp
$\dot{W}_T$	56195	56195	56195	56195	hp
$\dot{W}_{NET}$	20865	20865	20865	20865	hp
$\dot{Q}_H$	46939.2	46939.2	46939.2	46939.2	BTU/SEC
η_{TH}	0.3142	0.3142	0.3142	0.3142	

GAS-TURBINE POWER

TABLE OF RESULTS

ALLISON

P_1	14.48	PSIA
P_2	185.67	PSIA
P_3	178.24	PSIA
P_4	15.06	PSIA
T_0	100	°F
T_1	100	°F
h_1	133.55	$\frac{BTU}{LBm}$
P_{r1}	1.5712	
T_3	1700	°F
h_3	561.99	$\frac{BTU}{LBm}$
P_{r2}	20.1427	
h_{2is}	278.48	$\frac{BTU}{LBm}$
h_2	304.05	$\frac{BTU}{LBm}$
P_{r4}	22.42	
h_{4is}	286.93	$\frac{BTU}{LBm}$
h_4	314.44	$\frac{BTU}{LBm}$
$\dot{m}_a$	29.90	$\frac{LBm}{SEC}$
$\dot{m}_f$	0.4806	$\frac{LBm}{SEC}$
$\dot{W}_c$	7211	hp

GAS-TURBINE POWER

TABLE OF RESULTS

ALLISON

P_{4b}	6	6	6	6	PSIA
P_{5b}	5.76	5.76	5.76	5.76	PSIA
PR_{4-5}	2.615	2.615	2.615	2.615	
P_{r4}	8.934	8.934	8.934	8.934	
h_{4bis}	221.6	221.6	221.6	221.6	$\frac{BTU}{LBm}$
h_{4b}					$\frac{BTU}{LBm}$
ΔT		400	500		
T_{5b}		183.3	83.3		°F
h_{5b}		155.6	131.2		$\frac{BTU}{LBm}$
P_{r5b}		2.558	1.406		
P_{r6b}		6.688	3.675		
h_{6bis}		204.3	172.5		$\frac{BTU}{LBm}$
h_{6b}		212.9	179.7		$\frac{BTU}{LBm}$
$\dot{W}_{IDF}$		2462.4	2083.5		hp
$\dot{W}_{T}$		13162.5	13162.5		hp
$\dot{W}_{NET}$		3488.9	3867.8		hp
$\dot{Q}_{H}$		8613.0	8613.0		$\frac{BTU}{SEC}$
η_{TH}		0.2864	0.3175		

GAS-TURBINE POWER

TABLE OF RESULTS

ALLISON

P_{4b}	8	8	8	8	PSIA
P_{5b}	7.68	7.68	7.68	7.68	PSIA
PR_{4-5}	1.961	1.961	1.961	1.961	
P_{r4}	11.912	11.912	11.912	11.912	
h_{4bis}	240.38	240.38	240.38	240.38	$\frac{BTU}{LBm}$
h_{4b}	272.54	272.54	272.54	272.54	$\frac{BTU}{LBm}$
ΔT		400	500		
T_{5b}		248.8	148.8		°F
h_{5b}		171.36	147.16		$\frac{BTU}{LBm}$
P_{r5b}		3.621	2.101		
P_{r6b}		7.101	4.120		
h_{6bis}		207.77	178.13		$\frac{BTU}{LBm}$
h_{6b}		214.19	183.59		$\frac{BTU}{LBm}$
$\dot{W}_{IDF}$		1840.8	1565.9		hp
$\dot{W}_{T}$		12438.8	12438.8		hp
$\dot{W}_{NET}$		3386.8	3386.8		hp
$\dot{Q}_{H}$		8613.0	8633.0		$\frac{BTU}{SEC}$
η_{TH}		0.2780	0.3175		

GAS-TURBINE POWER

TABLE OF RESULTS

ALLISON

P_{4b}	10	10	10	10	PSIA
P_{5b}	9.6	9.6	9.6	9.6	PSIA
PR_{4-5}	1.569	1.569	1.569	1.569	
P_{r4}	14.89	14.89	14.89	14.89	
h_{4bis}	255.95	255.95	255.95	255.95	$\frac{BTU}{LBm}$
h_{4b}	286.55	286.55	286.55	286.55	$\frac{BTU}{LBm}$
ΔT		400	500		
T_{5b}		302.8	202.8		°F
h_{5b}		185.06	160.38		$\frac{BTU}{LBm}$
P_{r5b}		4.716	2.842		
P_{r6b}		7.398	4.458		
h_{6bis}		210.19	182.14		$\frac{BTU}{LBm}$
h_{6b}		214.67	185.98		$\frac{BTU}{LBm}$
$\dot{W}_{IDF}$		1270.5	1100.2		hp
$\dot{W}_{T}$		11836.7	11836.7		hp
$\dot{W}_{NET}$		3355.1	3525.3		hp
$\dot{Q}_{H}$		8613	8613		$\frac{BTU}{SEC}$
η_{TH}		0.2754	0.2843		

GAS-TURBINE POWER

TABLE OF RESULTS

ALLISON

P_{4b}	12	12	12	12	PSIA
P_{5b}	11.52	11.52	11.52	11.52	PSIA
PR_{6-5}	1.307	1.307	1.307	1.307	
P_{r4}	17.868	17.868	17.868	17.868	
$h_{4b_{is}}$	271.52	271.52	271.52	271.52	BTU/LBm
h_{4b}	300.56	300.56	300.56	300.56	BTU/LBm
ΔT		400	500		
T_{5b}		356.8	256.8		°F
h_{5b}		198.76	173.6		BTU/LBm
P_{r5b}		5.811	3.583		
P_{r6b}		7.695	4.796		
$h_{6b_{is}}$		212.61	186.15		BTU/LBm
h_{6b}		215.05	188.37		BTU/LBm
$\dot{W}_{IDF}$		700.2	634.5		hp
$\dot{W}_{T}$		11234.6	11234.6		hp
$\dot{W}_{NET}$		3323.4	3388.9		hp
$\dot{Q}_{H}$		8613	8613		BTU/SEC
η_{TH}		0.2728	0.2782		

APPENDIX B

CALCULATIONS FOR STEAM-TURBINE CYCLE

SOLUTION FOR T_x

1. $\dot{m}_t c_p (T_x - T_5) = \dot{m}_s (h_L - h_B)$

2. $\dot{m}_t c_p (T_4 - T_x) = \dot{m}_s (h_C - h_L)$

Divide 1./2. :

3. $$\frac{T_x - T_5}{T_4 - T_x} = \frac{h_L - h_B}{h_C - h_L}$$

Solve for T_x:

$$T_x = \frac{(h_L - h_B)}{(h_C - h_L)} \cdot (T_4 - T_x) + T_5$$

$$T_x = \frac{\dfrac{(h_L - h_B)}{(h_C - h_L)} \cdot (T_4) + T_5}{1 + \dfrac{(h_L - h_B)}{(h_C - h_L)}}$$

Subscripts refer to Figures 4 & 5.

STEAM-TURBINE POWER

T_C	$T_4 - 50$
h_C	Steam Tables [14] using P_B & T_C
h_L	Steam Tables using P_B and sat. liq.
h_B	$h_A + \bar{v}(P_B - P_A)$
T_x	From equation 4, previous page.
T_L	Steam Tables using P_B
$T_x - T_L$	Greater than zero?
$S_{d_{is}}$	S_c, Steam Tables using P_B
x_{is}	$S_{d_{is}}$, Steam Tables using P_A
$h_{d_{is}}$	x_{is}, Steam Tables using P_A
h_D	$h_c - \eta_{is_t}(h_c - h_{d_{is}})$
$\dot{m}_s$	$\dot{m}_t(h_4 - h_5) / (h_c - h_B)$
hp	$\dot{m}_s(h_c - h_D - h_B + h_A)$

STEAM-TURBINE POWER

TABLE OF RESULTS

TURBINE-EXHAUST PRESSURE = ATMOSPHERIC

ΔT	500	600	650	700	
T_c	1032.5	1032.5	1032.5	1032.5	°R
h_c	1535.6	1535.6	1535.6	1535.6	BTU/LBm
h_L	471.7	471.7	471.7	471.7	BTU/LBm
$h_c - h_L$	1063.9	1063.9	1063.9	1063.9	BTU/LBm
$\bar{v}(P_B - P_A)$	1.75	1.75	1.75	1.75	BTU/LBm
h_B	69.75	69.75	69.75	69.75	BTU/LBm
$h_L - h_B$	401.95	401.95	401.95	401.95	BTU/LBm
T_x	1179.6	1109.0	1070.7	1034.5	°R
T_L	946.3	946.3	946.3	946.3	°R
$T_x - T_L$	233.3	162.7	124.4	88.1	
T_5	1042.5	942.5	892.5	842.5	°R
s_c	1.7275	1.7275	1.7275	1.7275	BTU/LBm
x_{is}	0.8622	0.8622	0.8622	0.8622	
h_{Dis}	962.1	962.1	962.1	962.1	BTU/LBm
h_D	1076.7	1076.7	1076.7	1076.7	BTU/LBm
$\dot{m}_s$	11.61	13.84	14.94	16.05	LBm/SEC
$\dot{W}_{NET}$	7507	8949	9661	10379	hp

STEAM-TURBINE POWER

TABLE OF RESULTS

TURBINE-EXHAUST PRESSURE = 12 PSIA

ΔT	500	600	650	700	
T_c	955.9	955.9	955.9	955.9	°R
h_c	1493.6	1493.6	1493.6	1493.6	$\frac{BTU}{LBm}$
h_L	471.7	471.7	471.7	471.7	$\frac{BTU}{LBm}$
$h_c - h_L$	1021.9	1021.9	1021.9	1021.9	$\frac{BTU}{LBm}$
$\bar{v}(P_B - P_A)$	1.75	1.75	1.75	1.75	$\frac{BTU}{LBm}$
h_B	69.75	69.75	69.75	69.75	$\frac{BTU}{LBm}$
$h_L - h_B$	401.95	401.95	401.95	401.95	$\frac{BTU}{LBm}$
T_x	1107.1	1035.3	999.4	963.5	°R
T_L	946.3	946.3	946.3	946.3	°R
$T_x - T_L$	160.7	89.0	53.1	17.2	
T_5	965.9	865.9	815.9	765.9	°R
S_c	1.6987	1.6987	1.6987	1.6987	$\frac{BTU}{LBm}$
X_{is}	0.8467	0.8467	0.8467	0.8467	
h_{Dis}	945.9	945.9	945.9	945.9	$\frac{BTU}{LBm}$
h_D	1055.4	1055.4	1055.4	1055.4	$\frac{BTU}{LBm}$
$\dot{m}_s$	11.836	14.113	15.240	16.360	$\frac{LBm}{SEC}$
$\dot{W}_{NET}$	7306	8712	9408	10101	hp

STEAM-TURBINE POWER

TABLE OF RESULTS

TURBINE-EXHAUST PRESSURE = 10 PSIA

ΔT	500	600	650	700	
T_c	906.5	906.5	906.5	906.5	°R
h_c	1466.5	1466.5	1466.5	1466.5	$\frac{BTU}{LBm}$
h_L	471.7	471.7	471.7	471.7	$\frac{BTU}{LBm}$
$h_c - h_L$	994.8	994.8	994.8	994.8	$\frac{BTU}{LBm}$
$\bar{v}(P_B - P_A)$	1.75	1.75	1.75	1.75	$\frac{BTU}{LBm}$
h_B	69.75	69.75	69.75	69.75	$\frac{BTU}{LBm}$
$h_L - h_B$	401.95	401.95	401.95	401.95	$\frac{BTU}{LBm}$
T_x	1060.4	989.1	953.5	917.9	°R
T_L	946.3	946.3	946.3	946.3	°R
$T_x - T_L$	114.1	42.8	7.2	− 28.4	
T_5	916.5	816.5	766.5	716.5	°R
S_c	1.679	1.679	1.679	1.679	$\frac{BTU}{LBm}$
X_{is}	0.8362	0.8362	0.8362	0.8362	
h_{Dis}	935.03	935.03	935.03	935.03	$\frac{BTU}{LBm}$
h_D	1041.32	1041.32	1041.32	1041.32	$\frac{BTU}{LBm}$
$\dot{m}_s$	11.99	14.29	15.44	16.58	$\frac{LBm}{SEC}$
$\dot{W}_{NET}$	7181.8	8657	9248	9929	hp

STEAM-TURBINE POWER

TABLE OF RESULTS

TURBINE-EXHAUST PRESSURE = 8 PSIA

ΔT	500	600	650	700	
T_c	848.9	848.9	848.9	848.9	°R
h_c	1440.3	1440.3	1440.3	1440.3	$\frac{BTU}{LBm}$
h_L	471.7	471.7	471.7	471.7	$\frac{BTU}{LBm}$
$h_c - h_L$	968.6	968.6	968.6	968.6	$\frac{BTU}{LBm}$
$\bar{v}(P_B - P_A)$	1.75	1.75	1.75	1.75	$\frac{BTU}{LBm}$
h_B	69.75	69.75	69.75	69.75	$\frac{BTU}{LBm}$
$h_L - h_B$	401.95	401.95	401.95	401.95	$\frac{BTU}{LBm}$
T_x	1005.5	934.8	899.5	864.2	°R
T_L	946.3	946.3	946.3	946.3	°R
$T_x - T_L$	59.2	- 11.5	- 46.8	- 82.1	
T_5	858.9	758.9	708.9	658.9	°R
S_c	1.6554	1.6554	1.6554	1.6554	$\frac{BTU}{LBm}$
X_{is}	0.8233	0.8233	0.8233	0.8233	
h_{Dis}	921.7	921.7	921.7	921.7	$\frac{BTU}{LBm}$
h_D	1025.4	1025.4	1025.4	1025.4	$\frac{BTU}{LBm}$
$\dot{m}_s$	12.1	14.46	15.63	16.78	$\frac{LBm}{SEC}$
$\dot{W}_{NET}$	7071	8450	9134	9806	hp

STEAM-TURBINE POWER

TABLE OF RESULTS

TURBINE-EXHAUST PRESSURE = 6 PSIA

ΔT	500	600	650	700	
T_c	778.3	778.3	778.3	778.3	°R
h_c	1395.3	1395.3	1395.3	1395.3	BTU/LBm
h_L	471.7	471.7	471.7	471.7	BTU/LBm
$h_c - h_L$	923.6	923.6	923.6	923.6	BTU/LBm
$\bar{v}\,(P_B - P_A)$	1.75	1.75	1.75	1.75	BTU/LBm
h_B	69.75	69.75	69.75	69.75	BTU/LBm
$h_L - h_B$	401.95	401.95	401.95	401.95	BTU/LBm
T_x	940.0	870.2	835.5	800.6	°R
T_L	946.3	946.3	946.3	946.3	°R
$T_x - T_L$	−6.3	−76.1	−110.8	−145.7	
T_S	788.3	688.3	638.3	588.3	°R
S_c	1.624	1.624	1.624	1.624	BTU/LBm
X_{is}	0.8064	0.8064	0.8064	0.8064	
h_{Dis}	904.14	904.14	904.14	904.14	BTU/LBm
h_D	1002.4	1002.4	1002.4	1002.4	BTU/LBm
$\dot{m}_s$	12.42	14.82	16.02	17.21	LBm/SEC
$\dot{W}_{NET}$	6872	8,199	8863	9522	hp

APPENDIX C

CALCULATIONS FOR INDUCED-DRAFT FAN DESIGN

COMPRESSOR WORK

1.	T_{02}	$T_{01}(PR)^{\frac{\gamma-1}{\gamma}\frac{1}{\eta_p}}$
2.	η_p	Assume a value for iteration
3.	Δh_o	$c_p(T_{02} - T_{01})$
		Specify # Stages
4.	u	$(\Delta h_o$ / # Stages$)/ C_\theta$
	C_θ	$0.2369u$
	u^2	$(\Delta h_o$ / # Stages$)$ / (0.2369)
4.	u	$\sqrt{u^2}$
5.	C_2	$0.4649u$
6.	C_3	$0.742\ C_2$
7.	a_3	K&K [13] Table 2 using T_{03}
8.	a*	$a_3\left(\frac{\gamma+1}{2}\right)^{\frac{1}{2}}$
9.	M*	C_3 / a*
10.	P/P_o	K&K Table 30
11.	P_3	$P_{03}\ (P/P_o)$
12.	n	$\dfrac{\ln(P_3 / P_{01})}{\ln \frac{T_{01}}{T_{03}}\frac{P_3}{P_{01}}}$
13.	η_p	$\dfrac{(\gamma - 1) / \gamma}{(n - 1) / n}$
		Check with assumed value
14.	W_1	$1.077u$
15.	a_1	K&K Table 2 using T_{01}
16.	a*	$a_1\left(\frac{\gamma+1}{2}\right)^{\frac{1}{2}}$

17.	M^*	W_1 / a^*
18.	M_1	K&K Table 30
	COMPRESSOR	SIZE AND SPEED
19.	ρ_{o1}	$P_{01} / (R \cdot T_{01})$
20.	A_1	$\dot{m}/(\rho_{01} \cdot C_1)$
21.		Specify $r_h/r_t = 0.75$
	r_{t1}^2	$A_1 / 0.4375\pi$
22.	r_{h1}	$0.75\ r_{t1}$
23.	r_m	$(r_t + r_h) / 2$
24.	ρ_{02}	$P_{02} / (R \cdot T_{02})$
25.	A_2	$A_1 \cdot \frac{C_1}{C_2} \cdot \frac{\rho_{01}}{\rho_{02}}$
26.	r_{t2}	$(A_2/\pi + 4r_m^2) / 4r_m$
27.	r_{h2}	$2r_m - r_{t2}$
28.		Specify $p/q = 0.45$ (Diffuser)
	A_3	$1.29\ A_2$
29.	r_{t3}	$(A_3/\pi + 4r_m^2) / 4r_m$
30.	r_{h3}	$2r_m - r_{t2}$
31.	ω	u / r_m
32.	N	$30\,\omega / \pi$

Subscripts refer to Figure 6

INDUCED-DRAFT FAN

TABLE OF RESULTS

# STGS	1	2	3	4	5	
T_{02}	1158	1158	1158	1158	1158	°R
η_p	0.88	0.88	0.88	0.88	0.88	
Δh_{0t}	52.98	52.98	52.98	52.98	52.98	$\frac{BTU}{LBM}$
u	2366.9	1673.7	1366.6	1183.5	1058.5	$\frac{FT}{SEC}$
C_2	1100.4	778.1	635.3	550.2	492.1	$\frac{FT}{SEC}$
C_3	371.0	262.3	214.2	185.5	165.9	$\frac{FT}{SEC}$
a_3	1730.3	1730.3	1730.3	1730.3	1730.3	$\frac{FT}{SEC}$
a^*	1895.4	1895.4	1895.4	1895.4	1895.4	$\frac{FT}{SEC}$
M^*	0.2903	0.2053	0.1676	0.1451	0.1298	
P/P_0	0.9518	0.9756	0.9837	0.9878	0.9902	
P_3	14.33	14.69	14.81	14.87	14.91	PSIA
n	1.486	1.483	1.482	1.481	1.481	
η_p	0.874	0.877	0.878	0.879	0.879	
W_1	2549.2	1802.6	1471.8	1274.6	1140.0	$\frac{FT}{SEC}$
a_1	1496.2	1496.2	1496.2	1496.2	1496.2	$\frac{FT}{SEC}$
a^*	1639.0	1639.0	1639.0	1639.0	1639.0	$\frac{FT}{SEC}$
M^*	1.5553	1.0997	0.8980	0.7777	0.6955	
M_1	1.838	1.015	0.881	0.749	0.662	

INDUCED-DRAFT FAN

TABLE OF RESULTS

#STGS	3	4	5	
ρ_{01}	0.02292	0.02292	0.02292	$\frac{LBm}{FT^3}$
A_1	2.4248	2.7999	3.1305	FT^2
R_{t1}	1.328	1.427	1.509	FT
R_{h1}	0.9962	1.071	1.132	FT
R_m	1.162	1.249	1.321	FT
ρ_{02}	0.03515	0.03515	0.03515	$\frac{LBm}{FT^3}$
A_2	1.360	1.5709	1.7563	FT^2
R_{t2}	1.255	1.349	1.426	FT
R_{h2}	1.069	1.149	1.215	FT
A_3	1.834	2.118	2.368	FT^2
R_{t3}	1.288	1.383	1.464	FT
R_{h3}	1.036	1.115	1.178	FT
ω	1176	947.6	801.6	$\frac{RAD}{SEC}$
N	11230	9048	7654	RPM

Thesis 165477
H7115 Holmes
An inverted Brayton cycle application to naval marine gas turbines.